PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

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Development Bears Fruit

A BOAST that his company kept up its engineering and development work during the depression is a feature of almost every recent article or speech coming from the head of a manufacturing company in the metal product field. It was a logical time to do such work, of course, because orders were off and production was down, and interference was consequently at a minimum. At the same time the possibility that old markets might never revive made it essential for many companies to develop new products and to cultivate new markets.

The development work is now beginning to bear fruit. Take the field of production equipment typified by the machine tool. In December the index of orders for machine tools hit a high mark that has never even been approached before, either in the war years or at the peak of the 1929 boom. The major reason for this upsurge was the desire to replace as much as possible of the obsolete equipment in shops. But who made it obsolete? The development engineers with their new designs developed during the dull days of the depression.

Consumption of electric power is exceeding all records. Why? Because of the development work done by the engineers in the plants where electrical machinery is made for the home and for industry. Incidentally, the manufacturers of domestic electrical appliances are thinking in sales volumes far beyond anything ever dreamed of in the late twenties before the depression.

The railroads are getting back their passenger business with new types of cars developed since 1930. The airlines are carrying more people than they ever did, and so are the bus lines — in planes and buses greatly improved by the product engineers of those industries. As for automobiles, everybody knows how much better the new models are than those that went before, and who is responsible for the improvement. Henry Ford has just staged a little party as his twenty-five millionth automobile came off the assembly line. Would it have been possible for him to celebrate such an achievement if it had not been for continuous, intensive development activity?

LABOR is demanding a thirty-hour week on the utterly unsound theory that existing jobs represent the limit of our capacity and must be divided among those who want them. It will be obvious to any intelligent engineer or economist that we can have a thirty-hour work week when we have developed our production equipment to the point where our producers can turn out more product at less cost in thirty hours than they now do in forty. And not a minute sooner, unless we are willing to risk another major depression. We believe that the development work now going on will make a thirty-hour week practicable, but we reiterate that it will be the fruit of such activity and not an outcome of political pressure.

Chemical and physical laboratories, essential departments of the modern steel foundries, assure accurate control and a resulting product possessing a uniformly high quality

CHOOSING the Material and

RAYMOND E. COLLIER
Secretary
Steel Founders' Society of America

IT COULD probably be convincingly demonstrated that perspective is the most indispensable attribute of the successful designer of machine parts and mechanical structures. Not the kind of perspective that we associate with the drafting board, but the exercise of judgment in the choice of fabricating media, materials of construction, and the practical considerations involved in selecting the one best way to build a given product.

It seems fair to state that a designer's mind must always be open and receptive—ready to adopt this or that method or material, whichever will serve best on most counts. There must be fluidity in his viewpoint; a willingness to accept that which experience has proven to be most satisfactory even though he may have a personal predisposition toward something else.

Every engineering construction job must be worked out in accordance with its individual requirements and limitations. Few if any generalizations can be drawn with respect to choice of methods and materials for even items within a more or less homogeneous classification. It is folly to say that steel castings are the answer to all or even most structural parts problems. It is equally idle to say that welded construction is the panacea for everything, or that forging are the ultimate solution.

Shall a part be made of cast steel, malleable or grey iron, welded structurals or of nonferrous metals? The individual application must determine which one of these materials and constructions will serve best.

Usually the selection of materials is a compromise. One material might impart more desirable ultimate strength, another a higher and more advantageous yield point, a third might possess more nearly the impact resistance desired. One might resist corrosion better; another abrasion; still another high or low temperatures. Each engineered product carries with it a series of optimum requirements. It is seldom a material satisfies all of them in a superlative degree. The choice should go to that material possessing the highest aggregate rating on all of the requirements. And this compromise must evaluate the degree to which each requirement is essential to the finished product. A nice balance must be struck.

Most Failures Attributed to Design

Cost should not usually enter into these calculations but, as a practical matter, very often does. If failure of the product would be a serious matter, cost should be left out of the reckoning.

Some years ago, when the more recent movement toward welded construction was initiated the proponents of this fabricating method realized the importance of wiping the slate clean, so to speak, in approaching the question of designing welded structures. If the product had formerly been a casting the tradi-

Tensile tests of standard coupons are made to make certain that the steel has the proper strength and ductility

it profitable to work with the welding shop attractive possibilities in improved satisfaction would unquestionably follow.

While generalizations are odious it might be said that, with a limited number of exceptions steel castings are indicated wherever heavy dynamic stresses are imposed at points where maximum rigidity is essential in preserving such critically stressed areas intact. Where violent vibrations and impacts must be sustained by an irregularly shaped piece or part, steel castings will usually stand up better over the long haul. In cases where it is impossible to heat-treat a critically stressed member fabricated by welding in order to remove the strains set up by the welding process, it is usually more advantageous to go to cast steel. There are probably a number of exceptions to these general desiderata which could be advanced but would probably have to be considered as simply proving that the rule is generally applicable.

The writer recently visited a company that uses both welding and casting in the building of structures required in the construction of their own line of heavy machinery. When asked why they were casting a large number of parts that would normally appear to be "naturals" for welded construction the response was that it was simpler and easier to construct patterns

Construction Method

tional form and arrangement was to be dismissed from mind, and the new design arrived at on the basis of the requirements and limitations of the welding process. That was sound policy. It is no less true with respect to designing a steel casting. The same freedom from precedent and previous usage should characterize steel casting design work.

It is a known fact that most of the limited number of failures of steel castings are due to improper design. This is not entirely the fault of the steel foundry. The buyer very often neglects to inform the foundry as to the exact application of the casting, which would give the foundry an opportunity to study the design with a view to suggesting changes in the interest of a more satisfactory product, capable of best meeting the various service requirements. Frequent conferences between customers' designers and the foundry engineers pay real dividends in the form of better service records for the product that is conceived and produced by close cooperation.

If a user is having trouble with failure of welded structures a re-study of the design is imperative. This is equally true with respect to castings. It may well be that a change in fabricating method is not indicated but rather a redesign for more scientific production by the same method theretofore used. If the casting buyer were to work with the steel foundryman as closely as the buyer of welded structures has found

Ordinary carbon steel may be exceptionally ductile as evidenced by this test coupon that has been bent almost double without a sign of fracture

Optical pyrometers are used to carefully check the pouring temperature of the steel

and throw these items into the foundry than to obtain special steel plates and shapes, construct suitable jigs and make others for the proper manufacture of a welded construction. The pattern would always be available for repeat orders or replacement parts, and the dimensions would not have to be worked out and checked from the blueprint all the way through to the final stage in the case of repeat business. In many of these cases the welded structure might have saved a little money by reason of only one or two such parts being required. In spite of this fact other considerations outweighed the cost phase. This situation is cited to show how many and ramified are the reasons which determine a producer's choice of structural method or material.

Advantages Offered by Steel Castings

There is really very little difference between rolled steel and cast steel. A mild cast steel, specially adapted for welding, possesses characteristics quite comparable to rolled steel. Both are cast metal. The rolled steel was originally cast in the form of an ingot and its grain structure was mechanically refined in the rolling process. Steel castings are cast-to-final-shape, their grain structure being refined by proper heat-treatment. Section for section, one material has about the same characteristics as the other.

Most engineers and designers are aware of the advantages offered by the welding process. Not so many realize the points in which cast steel excels. There is the matter of smooth, nicely rounded, streamlined contours. A pleasing appearance adds to the salability of a product. There is the matter of rigidity. Steel castings just do not deform easily, although they will usually bend before they break unless they happen to be of a very high tensile, low ductility composition and such analyses are seldom used when sudden heavy impacts are encountered and where minimum wall thicknesses or sections are imperative.

Steel castings are versatile. The designer can place metal where it is most needed. Designing for cast steel is like molding with clay. A heavy section can be placed here, a boss there; a cast-on hook here, a journal box there. By consulting the steel foundry engineers the theoretically ideal model can be rationalized to make possible the most satisfactory distribution of the cast metal consonant with good foundry practice.

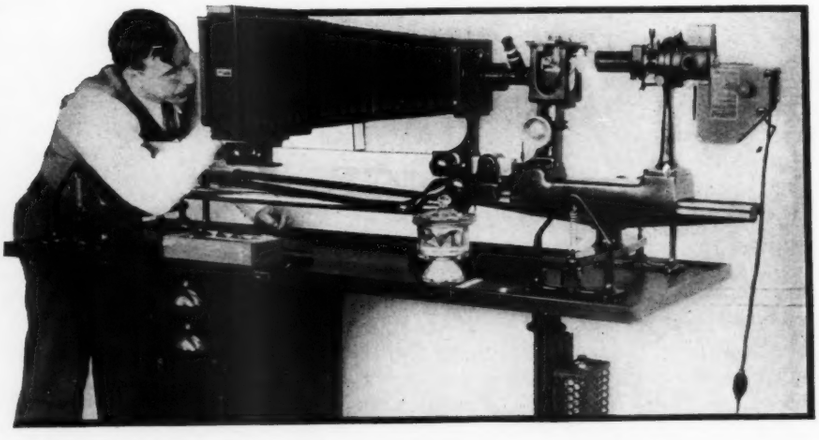
The value of the casting method for production jobs involving accurate alignment and close tolerances is well known.

It is important to note that an engineer may select any of several dozen alloy analyses when specifying a casting, if carbon steel is not as well suited to a specific application.

We conclude that the judicious choice of method and material is one which taxes to the utmost the disinterested open minded approach of the designer. He may list all of the vital considerations requisite to the production of the ideal or perfect product in the order of their importance and then rate the various alternative methods and materials, to arrive at a scientific solution. This may still not give the most satisfactory answer. But after all of the elements connected with the problem have been carefully weighed and evaluated, and the ultimate selection has been made, the designer should divorce himself from precedent and design his structure in a way best suited to the process and machinery that is going to do the production work. Changes from one method or material to another would probably seldom take place if this general procedure were followed.

The photographs used to illustrate this article indicate the careful inspection and control methods employed by a modern steel foundry so as to insure maximum customer satisfaction in connection with the products they supply.

In the above article Mr. R. L. Collier has emphasized the importance of an open mind in the selection of materials, and types of construction. Forthcoming articles in *Product Engineering* will summarize the considerations that enter into the selection of the type of construction and will enumerate the design requirements and possibilities associated with stampings, die-castings, sand castings, precision forgings, molded plastics, screw machine parts, pressure and impact extruded parts and also other constructions.



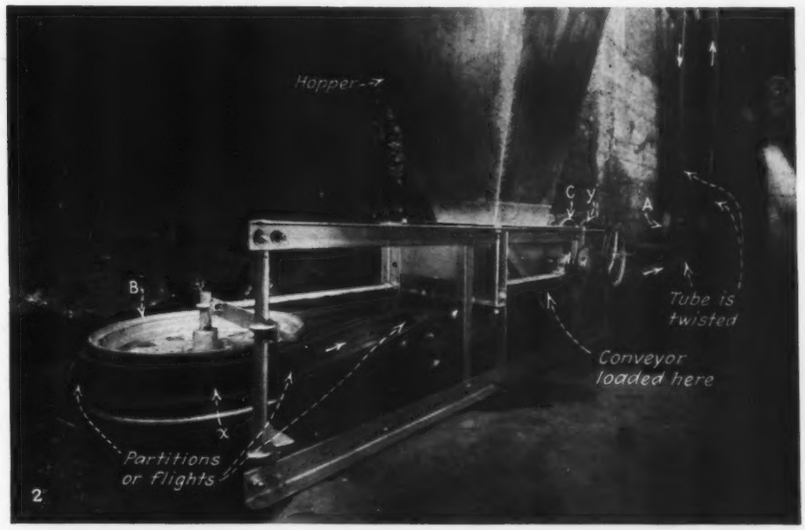
Modern scientific instruments such as this photomicrograph equipment have played an important part in the improvement of steel castings with respect to both their physical properties and their uniformity.

PRODUCT DEVELOPMENTS

Split Tubular Conveyor Made of Rubber

● A new type of conveyor recently announced by the Johns Conveyor Corporation is in effect a movable pipe line capable of travelling in any direction with as many changes of direction as desired. Made of rubber and reinforced with heavy chain for carrying the load, this separable conveyor is made endless in two half-sections. While conveying, the two halves are interlocked with double tongue and groove joints, as shown in Fig. 1. For unloading, the two halves are pulled apart. Wet or dry solids or liquids can be conveyed.

The lower half has disk-like partitions or flights at frequent intervals vulcanized to the rubber. These partitions assist in picking up a full load when passing under the feed hopper and also avoid slipping and packing during vertical lifting of the conveyed material. After loading, the halves are interlocked automatically, and the tube is then twisted to form a tight seal and to prevent separation. One of the features is that materials are conveyed without rubbing or tumbling, thereby causing a minimum of breakage and dust.



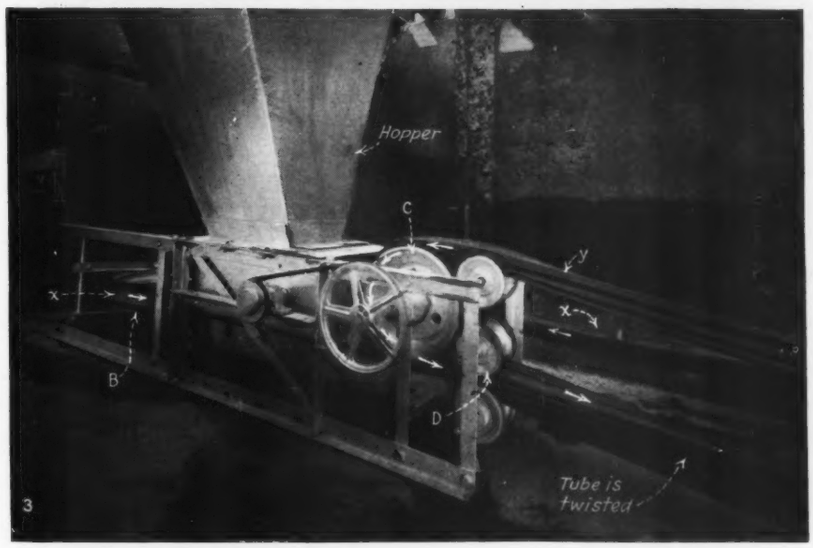
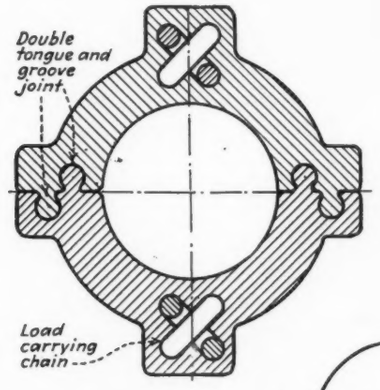


Fig. 2—Returning empty, the tube separates at idler pulley A, and the load-bearing half passes around horizontal pulley B to loading position under hopper. Fig. 3—Upper half of tube runs over pulley C joining lower half after loading between idler pulleys at D, after which the assembled tube is then twisted to seal the joint



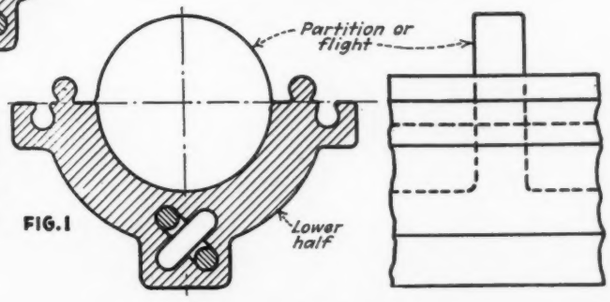
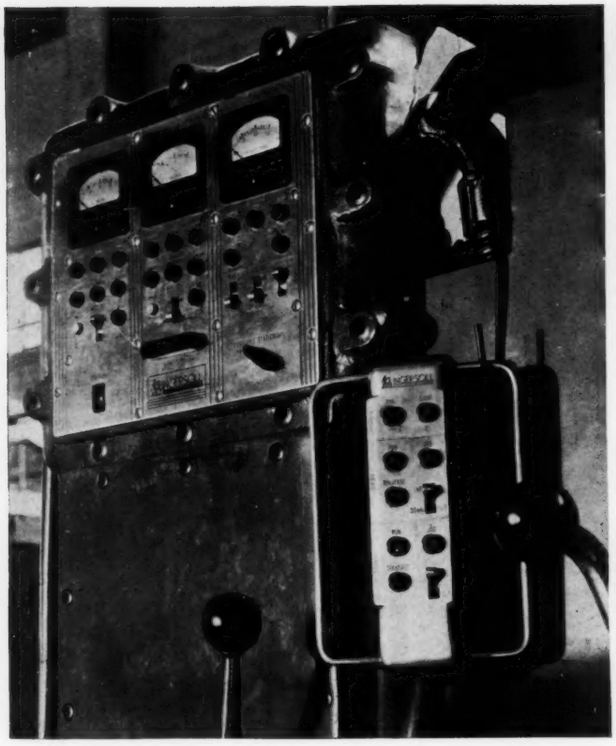
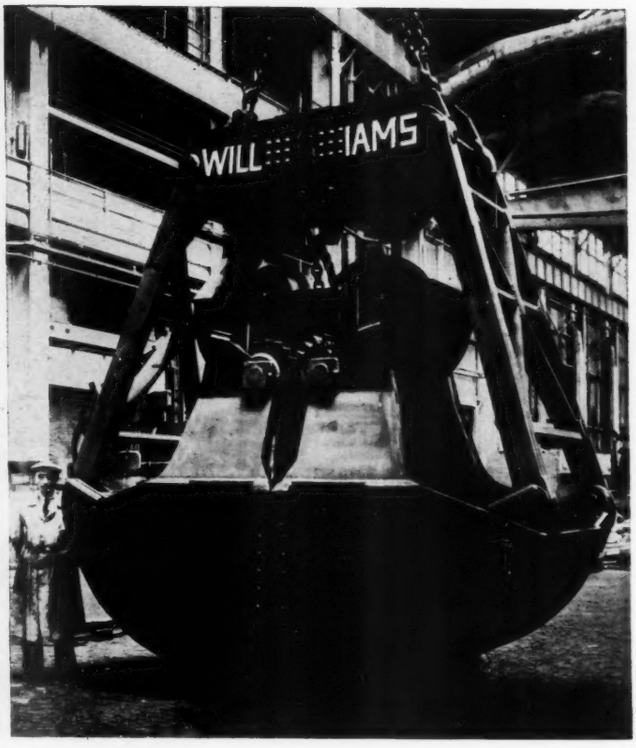


Fig. 1—Flexible tube conveyor made of rubber and reinforced with link chain, is joined and sealed with double tongues and grooves at each side

In the installation shown in Figs. 2 and 3 for conveying rice coal at the Manhattan Rubber Company plant, 9 tons per hr. are handled with a 2-in. internal diameter tube travelling 300 ft. per minute. As noted in Fig. 2 the return line at the left passes (empty) under pulley *A* with both halves joined. After passing this pulley, the halves are pulled apart, each half being guided around a different pulley. Because of the position of the loading station (hopper) in the line,





Built-in and portable electrical controls on the Ingersoll combination milling, drilling and boring machine. The three instruments indicate spindle speed, table travel and head travel

Coal bucket of 8½ cu. yd. capacity is fabricated entirely of nickel-copper-alloy steel by riveting and welding

the lower or loading half *X* of the conveyor is usually longer than the upper half. The lower half passes around pulley *B* in Figs. 2 and 3 and an idler pulley guides it open side up under the hopper for loading. At this point, the tongues and grooves are guided and kept clean to assure proper alignment and sealing. Meanwhile the upper half *Y* goes over pulley *C*, joining the lower half between the idler pulleys at *D* in Fig. 3. Immediately after leaving these idler pulleys, the assembled tube assumes a twisted form because of a twist which is imparted when the conveyor is originally assembled.

In this particular installation the bunkers are located 80 ft. above the loading station. The same procedure takes place for unloading as for loading. A 3-hp. motor is connected by a speed-reducing V-belt drive to the conveyor driving pulley near the unloading station. All pulleys and idlers with exception of the driving sheave revolve on antifriction bearings mounted on stationary shafts. Take-ups are provided at necessary points to maintain the required tension in the tube. Because of the light weight of the tube, the supporting structure is comparatively light. Should it be necessary to lengthen the conveyor while in the field, a section can be inserted.

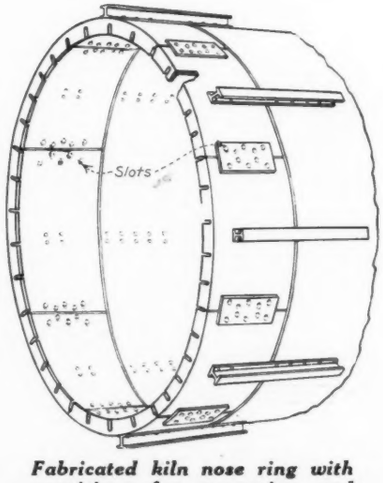
• *Built-In and Portable Electrical Controls*

A complete electrical control panel is built into the spindle head of the Ingersoll table type milling, drilling and boring machine, approximately at eye level when the operator is standing on the bridge in front of the cutter spindle head. This panel contains 26 pushbuttons and toggle switches for controlling the various movements of the machine. In the upper part of the panel are three instruments, one for registering the r.p.m. of the cutter spindle, and the other two for indicating the travel in in. of the head and the table.

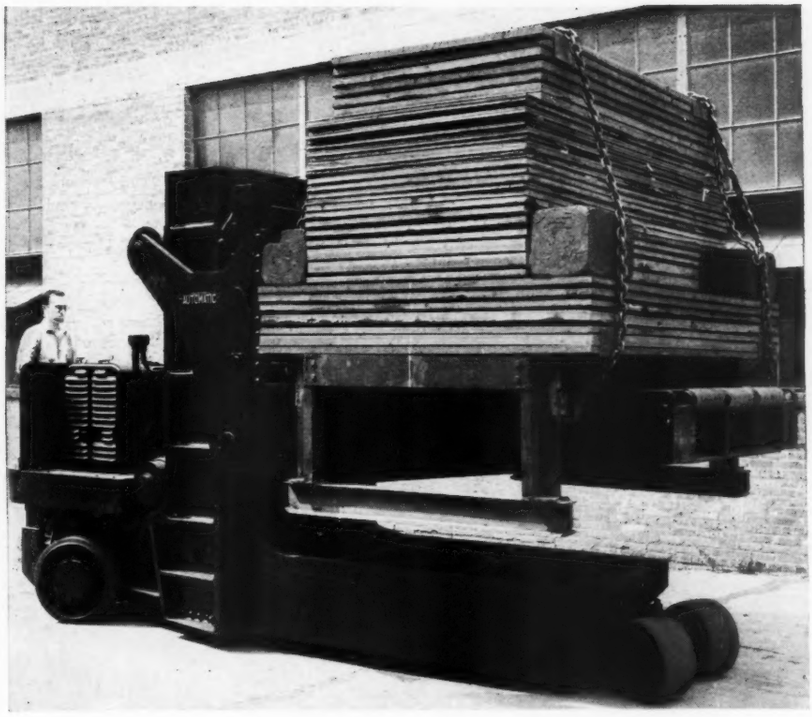
There is also provided a portable pendant switch, available with tachometer for reading spindle revolutions, which can be carried by the workman when operating the machine from the floor. When this pendant switch is used, a lever switch on the right of the panel board is thrown from "stationary" to "portable" and vice-versa when the pendant switch is again hung in the position shown.

• *Nickel-Copper for Impact and Corrosion Resistance*

Nickel-copper-alloy steel has been used throughout in the fabrication of the 7-ton capacity coal bucket constructed by The Wellman Engineering Company. Tests



Fabricated kiln nose ring with provisions for expansion and contraction through the use of segments and bolt slots



Tiering platform truck, built by Automatic Transportation Company for handling 30-ton dies to and from storage. Dual platform lift units, synchronized for equalized lifting, operate platform by four large double-strand roller chains. Drive unit has a V-joint universal for driving wheel connections. The eight-wheel trailing axle, fully equalized and compensating, is controlled by power steering

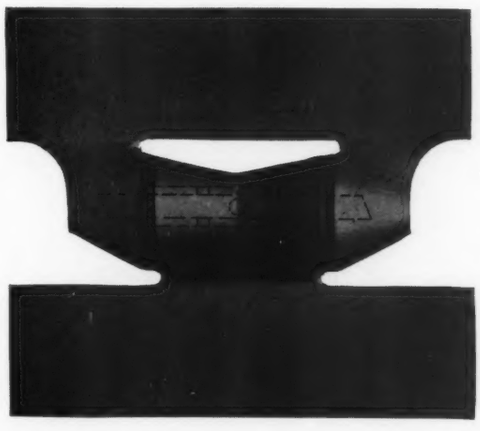
indicate that this alloy steel has a resistance to atmospheric corrosion of 4 to 6 times that of ordinary carbon steel and possesses high resistance to impact, both in the plate and at the welds. It has a yield strength of more than 60,000 lb. and an ultimate strength of approximately 90,000 lb. The jaws of this bucket have a 16-ft. reach, capable of scooping 8½ cu.yd. at one bite. The 18,000-lb. bucket is used in coal unloading service.

• *Preventing Distortion of Kiln Nose Ring*

Because of the damage to brick in rotary burning kilns caused by warpage and distortion of solid end rings, a special ring made up of eight manganese steel segments with provisions for expansion and contraction has been designed. The segments are held together by butt straps, bolted on through slotted holes to eliminate twisting. The whole assembly is supported from the main kiln shell by means of I-beams. The end flange of each segment is also slotted to provide for expansion or contraction. Manganese alloy steel is used because of its resistance to extreme temperatures and the corrosive action of sulphur gases developed during the burning operation.

• *Cast Hinge Is Water and Dust-Proof*

After frequent and prolonged use, hinges usually squeak, rust, or become filled with water or dust because of the imperfect fitting and excessive clearance between the hinge hub and pin. The accompanying illustration shows a hinge that may be die cast or sand cast with any metal or other material that has a lower melting point than the internal assembly of the hinge. It is built up of a round steel pin, the ends of which are slightly bevelled, a sleeve made of brass or bronze seamless tube mounted to rotate on the pin, and two pairs of washers of the same material located at the outer ends of this tubing. These pieces are anchored in place when the material is cast by flowing into holes in the washers and tubing walls. Both hinge leaves are cast at the same time.



Both leaves of the hinges are die or sand-cast simultaneously around pin, bushing and flanges

• *Random Jottings About New Developments*

For use on machine parts subject to wear from excessive abrasion and corrosion, exclusive of cutting tools, the Carboloy Company reports a sales increase of tungsten-carbide of 136 per cent for the first nine months of 1936 over the equivalent period in 1935, in this field. Among the parts on which Carboloy is used to overcome such wear are lathe centers, wire guides, valve stems and seats, textile guides, gages and others parts.

* * *

Among the new and original uses for stainless steel, as reported from England, are nameplates of stainless steel and brilliant finish that needs no cleaning. It is also reported that the locks and door fittings of stainless steel are being used throughout Marlborough House, one of the homes of Queen Mary.

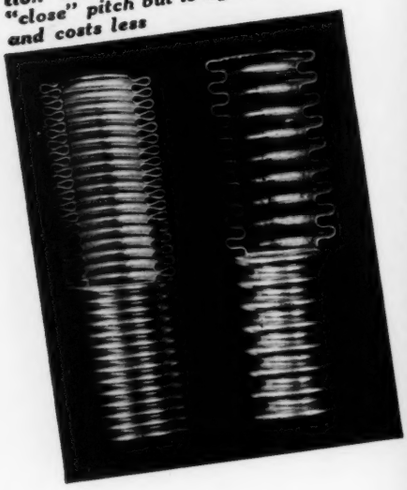
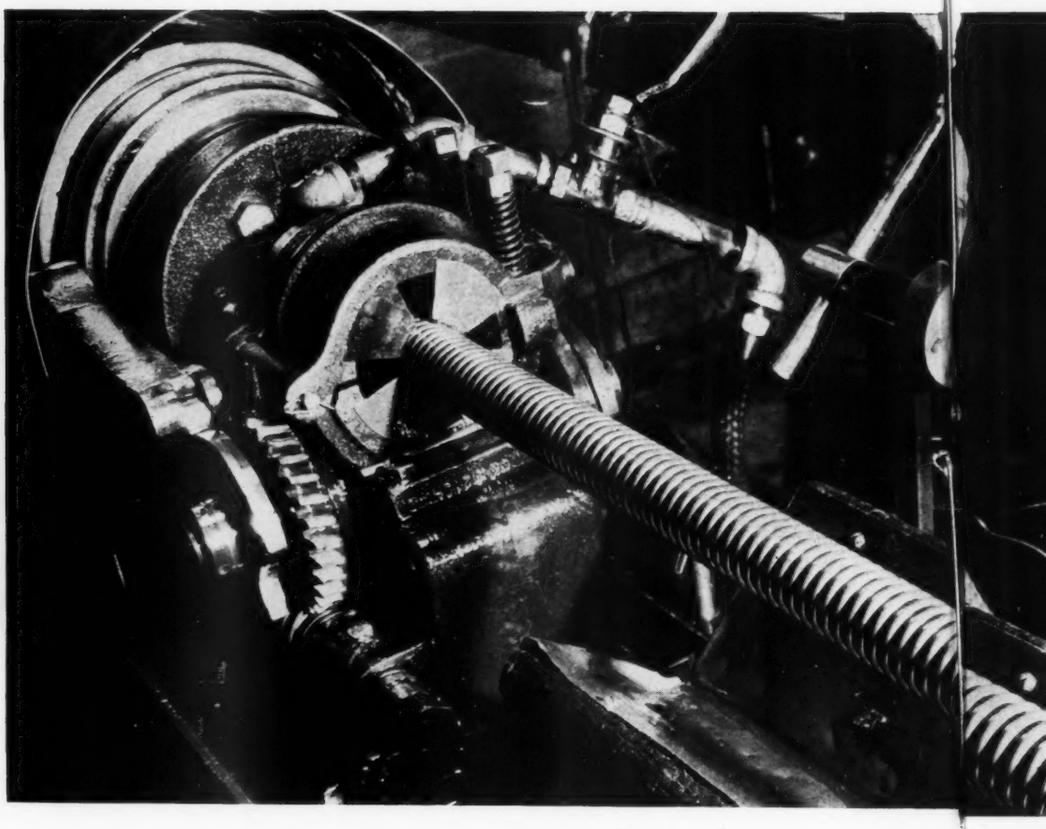


Fig. 1—Normal or "open" pitch tubing with straight-wall corrugation is not as flexible as the "close" pitch but is lighter per ft. and costs less



Seamless Flexible Metal Tubing

A. D. BLANCHARD
Eclipse Aviation Corporation

Types, sizes and end fittings available and how to install tubes to avoid failure

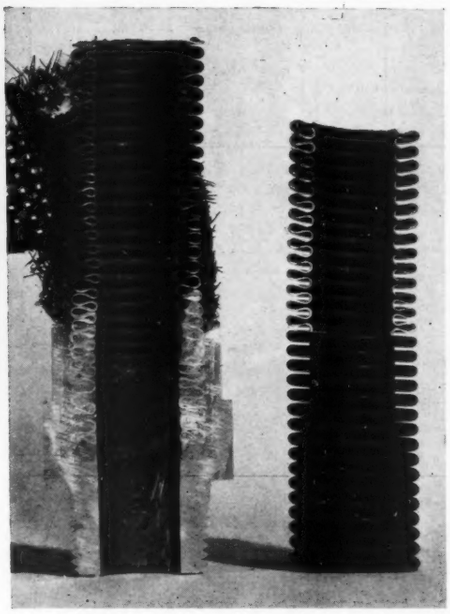
EACH type of flexible metallic hose is procurable in a great variety of designs and materials in order to meet the requirements of specific applications. Thus, the seamless corrugated type of metal hose, discussed in this article, is obtainable in various diameters, wall thicknesses, materials and pitches. Different types of armoring or braid coverings, and a great variety of end fittings are available. For best results, any given installation will require a certain combination of these many variables. It would be impossible to classify seamless metallic hose according to the varieties of design in which it is obtainable and the kinds of service for which each particular design is most suitable. The proper selection of the design of hose most suitable for a given application must be left to the engineers of the hose manufacturers, but knowledge of the various factors that enter into the problem will enable one to have a better understanding of the recommendations made.

The type of flexible metallic hose discussed in this article is made from thin-walled seamless tubing. By means of special machinery, deep corrugations of uniform depth and lying in parallel planes are made in the tube. The depth of the side walls of these corrugations is one of the factors from which the hose derives its flexibility. For a given diameter of tube, the other factors are the flexibility of the material, the thinness of the walls and the number of corrugations per unit length of tube.

As the flexibility of the corrugated tube is derived from the flexing of the side walls of the corrugations, it is evident that the greater the number of corrugations in a given length of tube the greater will be its flexibility. But the greater the number of corrugations the greater will be the amount of material required, increasing the cost and weight correspondingly. Therefore the corrugated tubing is manufactured with normal or open pitch for applications wherein extreme flexibility is not needed, and is made with close-pitched corrugations for applications wherein extreme flexibility is required. These two types are shown in Fig. 1, the normal pitch tubing having straight side walls and the close pitched tubing having curved side walls, the corrugations being in parallel planes in both designs. That

Fig. 2—Automatic machines transform the smooth seamless tubing into flexible metal hose by forming corrugations in the wall of the tube

Fig. 3—A hydraulic pressure of 2,000 lb. crush the walls of the sectioned tube to the left, the double covering of braid preventing it from bursting. The undeformed tube to the right is shown for comparison



is, the corrugations are not in the form of a helix,

From the nature of its design, no form of seamless flexible tubing can be made with heavy side walls and withstand an appreciable number and amount of flexings. The tube bends by virtue of the bending of the side walls of the corrugations. For a given amount of bending, the stresses developed in the walls will be proportional to the thickness of material. In addition, there will be the stresses set up by the internal pressure to which the tube is being subjected. If the total combined stresses thus developed are greater than the endurance limit of the material, the tube will fail by fatigue.

Evidently there are two conflicting requirements in the design of tubes. The wall thickness must be sufficient to withstand the internal pressure to which the tube will be subjected, but on the other hand the tube wall should be as thin as possible in order to minimize the stresses created by flexing. To eliminate one of these conflicting requirements, the tubes are designed solely for flexibility and fluid tightness. If the tube is to be used for conducting fluids under high pressure, it is provided with an armor in the form of a flexible woven wire covering or braiding. The tube then acts much in the same manner as the inner tube of an automobile tire. It is air and liquid tight but the bursting resistance is contributed practically entirely by the casing or armor.

In the accompanying table are given the permissible working pressures for plain and braid covered flexible

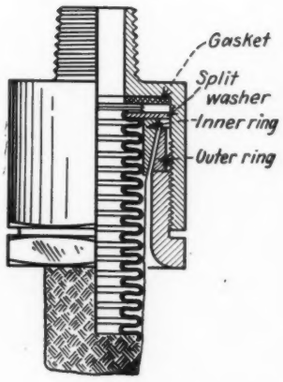
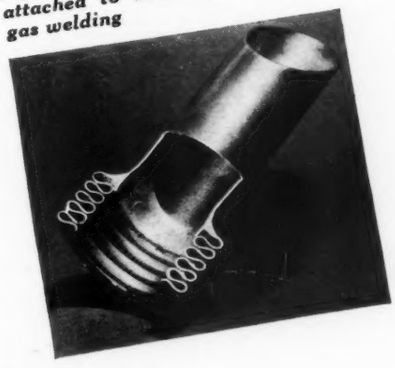


Fig. 4—Aluminum fittings are attached to aluminum hose by gas welding



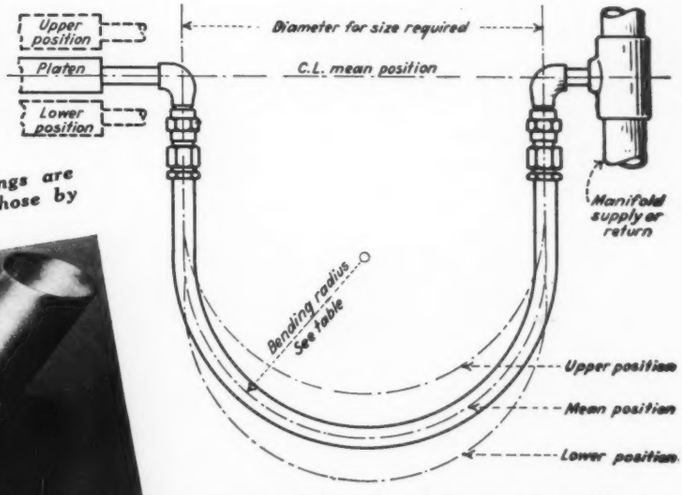


Fig. 5—Typical application of flexible tubing, indicating how to calculate the length of tubing required

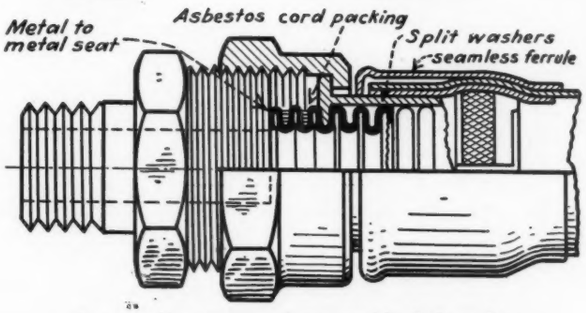


Fig. 6—Two types of renewable fittings for flexible tubing with parallel corrugations

corrugated tubing, together with other data. The figures given for rated pressures are conservative. The ratings will vary over a wide range depending mainly upon the amount of flexing to which the tube will be subjected. Also, materials other than the two covered by the table can be used for the tubes. Thus, Monel corrugated tubes will permit materially higher ratings than those given in the table. A 3 per cent silicon bronze will likewise permit higher working pressures, but not as great as permissible with Monel tubes.

The manner in which flexible corrugated tubes fail under excessive pressure is shown in Fig. 3. This is a section of a $\frac{3}{4}$-in. diameter close pitched tube that

SINGLE SEAMLESS HOSE DATA

Approximate Dimensions of Unarmored Hose					Approximate Weights lb. per ft.				Approx. Min. Bend. Radius Close Pitch	Rated Pressures (lb. per sq. in.)			
Inside Diameter In.	Outside Diameter In.	Standard Wall Thickness	Standard Pitch Per In.		85 Cu.-15 Zn		4S Alum.			Without Armor	With One Braid	With Two Braids	With Three Braids
			Normal	Close	Normal	Close	Normal	Close					
1/8	0.280	0.006	15	23	0.041	0.053	0.013	0.019	4	100	500	1000	3000
3/16	0.400	0.006	11	16.5	0.079	0.119	0.024	0.036	5	100	500	1000	3000
1/4	0.515	0.008	8.5	13	0.128	0.195	0.039	0.059	5	90	500	1000	3000
5/16	0.600	0.009	8	12	0.145	0.218	0.046	0.069	7	75	150	400	2000
3/8	0.692	0.010	7	11	0.195	0.306	0.059	0.093	7	60	150	400	1200
1/2	0.900	0.011	6	9	0.318	0.475	0.094	0.145	8	50	150	250	1200
9/16	0.978	0.013	5	7.5	.....	.....	.....	.....	9	50	150	250	500
5/8	1.060	0.013	5.3	8	0.398	0.605	0.122	0.183	9	50	150	250	500
3/4	1.240	0.014	4	6.5	0.482	0.71	0.155	0.230	10	30	150	250	500
13/16	1.375	0.016	3.5	6.2	.....	.....	.....	.....	..	....	....	....	....
1	1.600	0.016	3.8	5.7	0.67	1.0	0.237	0.339	12	30	150	250	400
1 1/8	1.760	0.016	3.6	5.5	0.775	1.2	0.236	0.360	13	30	150	250	350
1 1/4	1.940	0.017	3.4	5	0.932	1.37	0.284	0.417	13	30	150	250	350
1 1/2	2.260	0.018	3	4.6	1.165	1.635	0.325	0.498	14	25	150	225	300
2	2.920	0.018	2.5	3.8	1.495	2.27	0.453	0.690	17	20	125	200	250
+ 0.015 − 0.000	+ 0.005 − 0.005	Varies slightly depending on alloy	Varies slightly depending on alloys and manufacturing tolerances		Weights vary slightly due to variations in wall thickness, pitch, and alloy				Varies due to pressure, pitch and alloy	Working conditions control pressure for which hose is suitable. Braid prevents increase in length and gives required strength.			

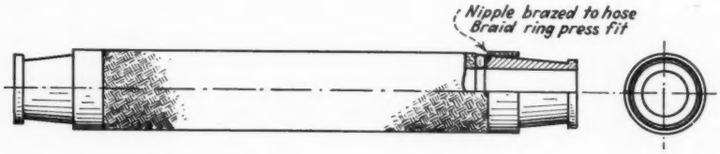


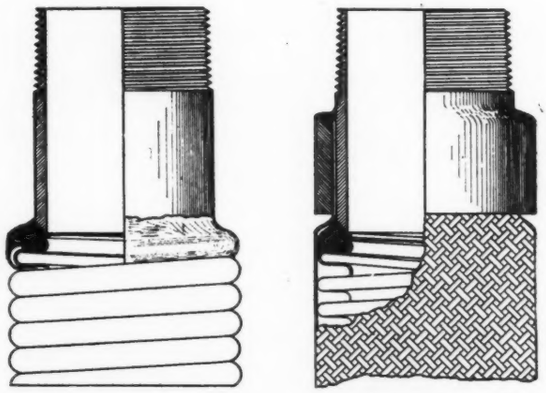
Fig. 7—Three examples of the great variety of permanent end fittings attached to the metal hose by brazing

has been subjected to a hydrostatic pressure of 2,000 lb. per sq. in. For comparison, a section of undistorted tubing is shown right of tested sample. It will be noticed that the bottom of each corrugation in the tested sample has been squeezed to a smaller radius, which was caused by the pressure on the side walls. Also, the internal pressure caused the top of each convolution to deform to a larger radius. It is apparent that such a distorted tube has lost much of its flexibility and that the metal has been stressed beyond its elastic limits.

Internal pressure causes any tube, be it a fabric fire hose or a metallic tube, to resist flexing. Definite figures are not available, therefore the designer must be guided by the engineers of the tube manufacturer in considering this factor. Sometimes it is possible to arrange the design so that the tube will not be required to flex while under pressure. For example, in the design of a press with platens heated by high-pressure steam it was arranged so that the pressure in the seamless flexible metal hose supplying steam to the platens was released during their travel. Thus the tube was not submitted to pressure during its flexing.

Corrugated flexible tubes cannot withstand simultaneous torsion and flexing. This would set up complex stresses that would soon cause fatigue failure. Therefore in designing a flexible tube installation care must be taken that the tube will lie in a plane and will remain in the same plane at every position of its travel. If the line of travel of the end connection is not in the plane of the suspended tube, the tube will be subjected to twist. To overcome such a condition, rigid piping should be used to shift the plane of suspension of the tube in order to bring it into the line of travel of the moving end.

The approximate minimum bending radius as given



in the tabulated data for close pitched tubing will vary according to the internal pressure to which the tube will be subjected and the alloy out of which it is made. The figures given can safely be used for preliminary design calculations. Higher pressures increase the minimum radius of curvature. Open pitched tubing likewise requires a greater minimum radius of curvature.

If the ends of a flexible tube are rigidly anchored, there will be an abrupt change of stress conditions at the tube sections adjacent to the anchorages. The anchored ends of the tube will not be able to flex to conform with the flexing of the free portions of the tube. Therefore the metal in the free portion of the tube close to the fittings will be bent sharply, thus causing high stress concentration. Continuous vibration under these conditions will cause fatigue failure, the tube breaking at one of the end fittings.

To eliminate the possibility of the fixed ends creating a condition of stress concentration, one or two extra reinforcing braid coverings are put on the ends

of the tube. These reinforcing coverings are about 6 in. long and are firmly anchored in the fitting. When two such coverings are used the outer one is somewhat shorter. In this manner there is a gradual increase in the rigidity of the tube as it approaches the end fitting and the possibility of stress concentration with subsequent fatigue failure will be eliminated.

A typical installation for seamless metal hose is shown in Fig. 5. The overall length of hose required is determined as follows: the base length is taken as twice the bending diameter which diameter must not be less than the minimum permissible radius of bend. To the base length is added the total movement of the moving end. This assumes that the fixed end is opposite the mean position of the moving end, as in Fig. 5. If the fixed end is above or below this mean position, that distance must be added to the length of hose required, measured from end to end of fittings. The same general rule applies for horizontal movements. Instead of using a number of different lengths, it is good practice to make all pieces the maximum length required so that they will be interchangeable.

In addition to 85 Cu-15 Zn alloy and 4S aluminum, corrugated seamless metal hose is made of 3 per cent silicon bronze, Monel, nickel and silver. The last two are used for special applications in the chemical industries. Silicon bronze hose is being widely used for oil burners and to conduct Freon and other refrigerants. Monel is being used for high pressure superheated steam and where strength is a controlling factor. Aluminum is used primarily for lightness as in aviation. The 85 Cu-15Zn alloy is suitable for saturated steam up to 250 lb. per sq. in. pressure, the corresponding saturation temperature being 406 deg. F. Sudden temperature changes such as when steam is followed by cold water have no unfavorable effects, although for this service Monel is recommended.

Braided coverings may be made of bronze, steel or any other metal. Stainless steel is frequently used for its corrosion resistant properties. Sometimes an additional armor of interlocked steel or bronze casing is added to protect the braids. The casing also makes it impossible to kink the hose. This is of special advantage if the hose is to receive much handling, in which case the interlocked steel casing also protects the hands of the operator from being injured by broken braids.

End fittings, of which there are a great variety, are either of the removable type or the permanently attached type. One advantage of the parallel corrugated tubing as compared to the helical corrugated tubing is that it permits a simplified design of renewable fittings. Two such designs of fittings are shown in Fig. 6. Their construction is self-evident. Permanently attached fittings for any kind of connection are brazed or soldered to the tubing. Typical designs are shown in Fig. 7. End fittings for aluminum hose, as shown in Fig. 4, are attached by welding.

Seamless flexible hose is practically always sold with the fittings attached. Special equipment and an expert technique is required in assembling the hose. The length of the hose is always specified from end to end of fitting. Special fittings made to sketches can be obtained.

Maximum single piece lengths of seamless corrugated tubing are approximately 10 ft. If a greater length is required it may be obtained by joining pieces by brazing, an example of which is shown in Fig. 8. The tubing will be stiff at the brazed joint but this is usually not detrimental in long lengths of tubing.

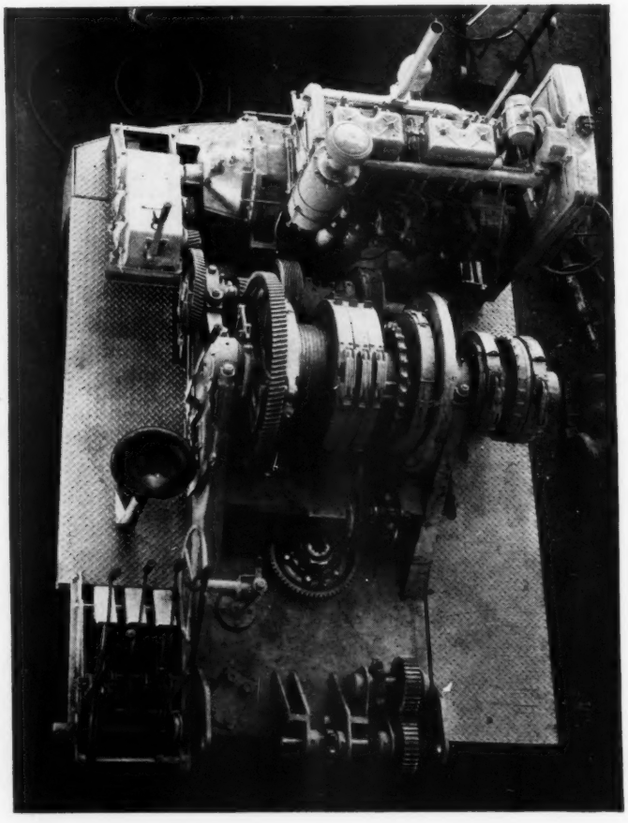
In addition to the applications already mentioned, seamless tubing is used for exhaust connections, carrying high pressure oil in hydraulically operated machines, for supplying lubricating oil to moving elements of a machine, and as a connection to isolate vibrations. In addition to these, there are innumerable special applications.

Fig. 9—Brazing the end fittings to the seamless flexible hose in the plant of the Eclipse Aviation Corporation

Fig. 8 — Design of brazed joint used for connecting sections to obtain a long length of tubing

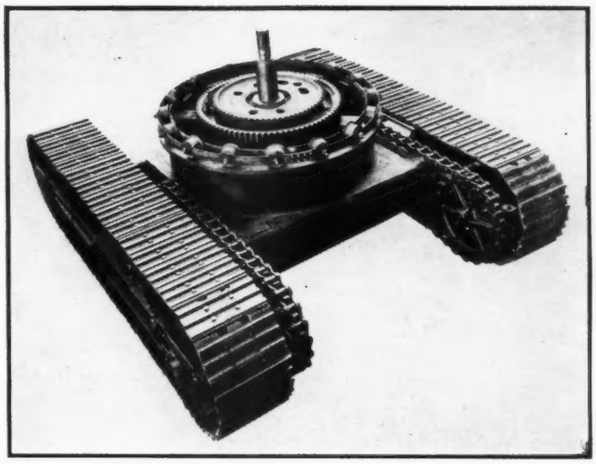
Modern Mechanical

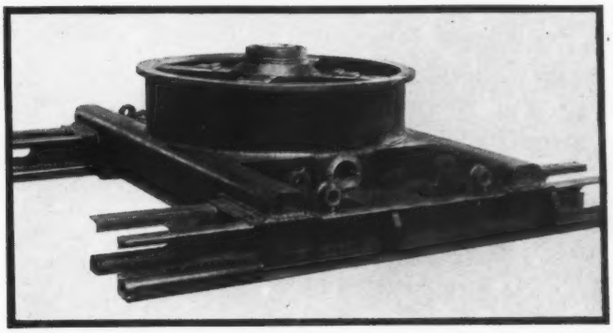
Light, strong and powerful, the 1 yd. P & H Pacemaker, a product of the Harnischfeger Corporation, gives high production by easy control and fast operation. Alloy steels, welding and quality design made it possible



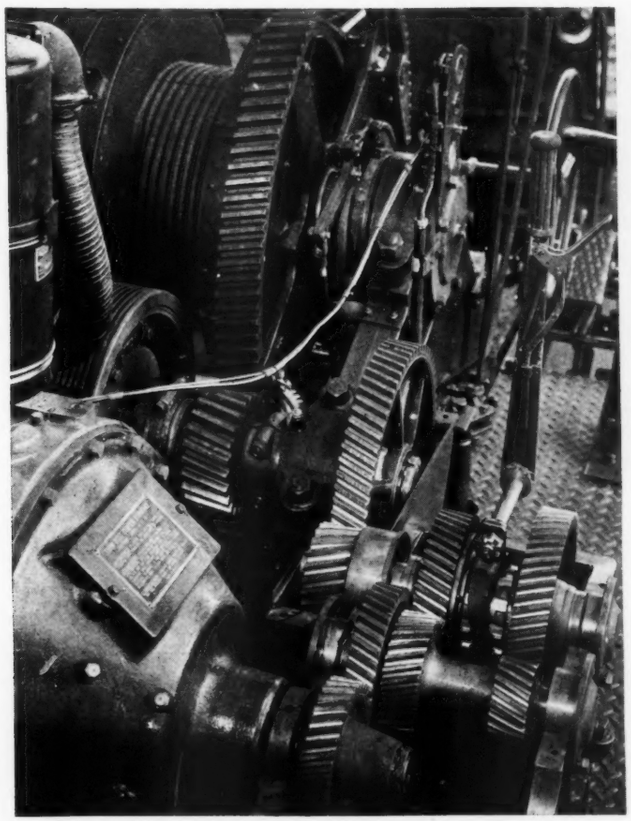
IN THE arrangement of the machinery on the deck, special care was taken to make all parts accessible without sacrificing compactness. All the heavy main units including the 4-cylinder Caterpillar diesel engine and the hoisting unit were placed behind the center of rotation in order to balance the front end without necessitating counter weights. Two-speed transmission, fully inclosed, is provided for every operating motion. All jack shafts are of chrome-vanadium steel, machined and ground; all high-speed shafts are mounted on anti-friction bearings. A centralized Alemite lubricating system insures proper lubrication at all points and greasing is the work of only a few minutes.

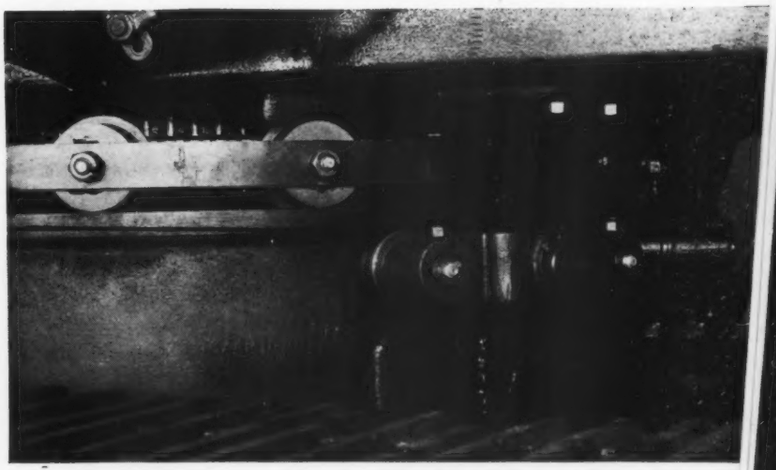
LOW-ALLOY high-strength steel is used throughout the arc-welded one-piece car body and crawler frame shown below. The elimination of castings and mechanical joints resulted in a much lighter structure but one that is more rigid, with greater than usual protection against weaving. In the crawler assembly to the left can be seen the heat-treated rolled steel shoes, shaped to throw off the dirt readily, making steering easier. Drive chain sprockets are heat-treated alloy steel, and the chain is made extra heavy. Compensator springs give the crawler track resiliency.





Designs—*The P & H Excavator*





UPPER structure rotates on live roller circle having 16 flanged rollers. To counteract tipping strains, thereby eliminating excessive pull on the center pin, double swiveled hook rollers are used at the back, contacting the lower flange of the roller track.

FOR smooth action and quiet operation, all gears in the hoist train are helical cut, made of alloy steel and designed so that there will be three teeth in mesh at all times. The gears are fully inclosed and run in an oil bath. Clutches are Twin Disc, adjusted for sensitive control.

Formulas for Deflections and Bending Moments

RAYMOND J. ROARK

Professor of Mechanics, University of Wisconsin

IN THE tables on the following pages are given formulas for the end reactions, end moments, vertical shears, bending moments, deflections, and end slopes of beams supported and loaded in various ways. The cases considered have been so selected that, by superposition, the formulas are applicable to almost any combination of loading and support likely to occur.

These tables differ from similar tables to be found in texts and handbooks only in respect to completeness; a number of cases of partial loading, not usually given, are included, as are also end slopes, useful in determining the effects of overhang and partial end constraint.

All formulas for statically indeterminate beams, and all deflection and slope formulas for statically determinate beams, are based on the following assumptions:

1. The beam is of homogeneous material having the same modulus of elasticity in tension and compression; it is straight, of uniform cross-section, and has a longitudinal plane of symmetry. If the beam does not have a longitudinal plane of symmetry, it will be subjected to a twisting moment by any load not passing through a certain point in the cross-section called the "flexural center," which point does not coincide with the centroid. A line containing the flexural centers of all sections is called the "elastic axis," and all loads and reactions must lie in a plane containing this elastic axis if torsion is to be avoided. (See "The Problem of Flexure and its Solution by the Soap Film Method,") by A. A. Griffith and G. I. Taylor, Reports and Memoranda No. 399, Advisory Committee for Aeronautics (British), Nov., 1917. See also "Remarks on the Elastic Axis of Shell Wings," by Paul Kuhn, Technical Note No. 562, National Advisory Committee for Aeronautics (U.S. April, 1936.)

(The position of the beam is immaterial, but for convenience it will be assumed horizontal.)

2. All loads and reactions are normal to the axis of the beam, and lie in the aforesaid plane of symmetry. (For convenience, this plane, and consequently all loads and reactions, will be assumed vertical.)

3. The supports are rigid with respect to vertical displacement.

4. The beam is not stressed beyond the limit of proportionality, and does not deflect excessively. Excessive deflection, for the present purpose, may be defined as any deflection that results in a maximum slope of 25 per cent or more.

5. The beam is so long in proportion to its depth that the deflection, because of shear, is negligible in comparison with deflection caused by bending. For metal beams of solid rectangular or circular section, deflection because of shear may usually be considered negligible when the ratio $l/d = 8$ of span to depth is exceeded. For metal I-beams and similar sections the effect of shear is greater, and the limiting ratio of 15 may be taken. For smaller values of l/d, formulas for deflection because of shear should be employed. (See "Strength of Materials," A. Morley, or "Strength of Materials," J. Case, or "Strength of Materials," S. Timoshenko.)

In timber beams the effect of shear is greater than in metal beams. For beams of rectangular section, having a ratio of l/d between 12 and 24, it may be allowed for by using for E a value 10 per cent less than the true value, or by using a value determined from bending tests on beams of similar proportions. If l/d exceeds 24, the true value of E should be used; if l/d is less than 12, or if the beam is an I-beam or box beam, the effect of shear should be computed. (See "Deflection of Beams with Special Reference to Shear Deformation," by J. A. Newlin and G. W. Trayer, Report No. 180, National Advisory Committee for Aeronautics U.S. 1924.)

Notation, Definitions and Units

The notation employed, definitions of terms, and units in which the quantities involved should be expressed, are as follows:

E is the modulus of elasticity in lb. per sq.in. of the material.

I is the moment of inertia (in.^{4}) of the cross-section of the beam about the neutral axis.

l is the length in in. of the beam between supports.

M is the bending moment in in.lb. at any section of the beam. It is the algebraic sum of the moments of all loads and reactions that act on the beam to the left of the given section, these moments being taken about the neutral axis of the given section and being considered positive when clockwise, negative when counter-clockwise.

M_1 and M_2 are the external constraining or fixing moments in in.lb. at the left and right ends respectively of the beam; they are couples that act as shown in the diagrams.

M_o is an external couple in in.lb. applied as a load; it is positive when clockwise, negative when counter-clockwise.

R_1 and R_2 are the vertical reactions in lb. acting at the left and right ends respectively of the beam; they are positive when upward and negative when downward.

V is the vertical shear in lb. at any given section of the beam. It is the algebraic sum of the vertical loads and reactions that act on the beam to the left of the given section, and is positive when acting upward, negative when downward.

W is the total load in lb. on the beam.

w is the maximum intensity of distributed loading in lb. per linear in. on the beam.

x is the horizontal distance in in. to any given section of the beam, measured from an origin taken at the original position of the left end of the beam.

y is the vertical deflection in in. of any point on the neutral surface of the beam from its original or unloaded position; it is positive when upward and negative when downward.

θ is the angle in radians between the horizontal and the axis of the loaded or deflected beam; it is positive when the beam slopes up to the right and negative when the beam slopes down to the right. Within the limitation here imposed as to slope, it may be assumed that $\theta = tan\ \theta = sin\ \theta$ and that $cos\ \theta = 1$.

$a,b,c,\ldots z$ denote distances in in. measured horizontally along the beam, as indicated in the diagrams.

$A,B,C,\ldots Z$ denote points on the axis of the beam, as indicated in the diagrams.

Examples

The use of the formulas to solve, by superposition, any case that may be regarded as a combination of several of those given in the tables, is illustrated by the following examples.

In Fig. 1, a 2 by 4-in. spruce beam 15 ft. long is supported at the right end and at 40 in. from the left end. It carries a concentrated load of 150 lb. at the left end and a load of 300 lb. uniformly distributed over the portion between supports. The 4-in. dimension is vertical. The modulus of elasticity is 1,300,000 lb. per sq.in. It is required to determine the deflection of the left end.

The deflection y at the left end is the sum of the upward deflection y_1 produced by the distributed load and the downward deflection y_2 produced by the concentrated load. Each of these deflections is computed independently of the other, thus:

$$y_1 = -40\ \theta$$

By the formula for end slope as in Case 13,

$$\theta = -\frac{1}{24}\frac{(300)\ (140^2)}{EI} = -\frac{245{,}000}{EI}$$

$$\text{and } y_1 = -40\left(\frac{-245{,}000}{EI}\right) = +\frac{9{,}800{,}000}{EI}$$

By the formula for end deflection as in Case 1,

$$y_2 = -\tfrac{1}{3}\frac{(150)\ (40^3)}{EI} = -\frac{3{,}200{,}000}{EI}$$

$$\text{Hence } y = \frac{9{,}800{,}000 - 3{,}200{,}000}{EI} = +\frac{6{,}600{,}000}{EI}$$

Substituting 1,300,000 for E and $[(1/12)(2)(4^3)] = 10.67$ for I, it is found that $y = +0.47$ in.

In Fig. 2, the 6-in., 12.5 lb. I-beam 20 ft. long is fixed at the right end, supported at 5 ft. from the left end, and carries, in addition to its own weight, the loads shown. It is required to determine the reaction at the left support.

The beam, exclusive of the overhanging portion, is regarded as a beam fixed at the right end, supported at the left end, and loaded as shown in the lower diagram, Fig. 2, with a uniform load equal to its own weight, the trapezoidal load, a load at the left end equal to the shear imposed by the overhanging portion, and a couple at the left end equal to the bending moment imposed by the overhanging portion. This shear and couple are readily calculated, and their values are as given. For convenience, the several loads will

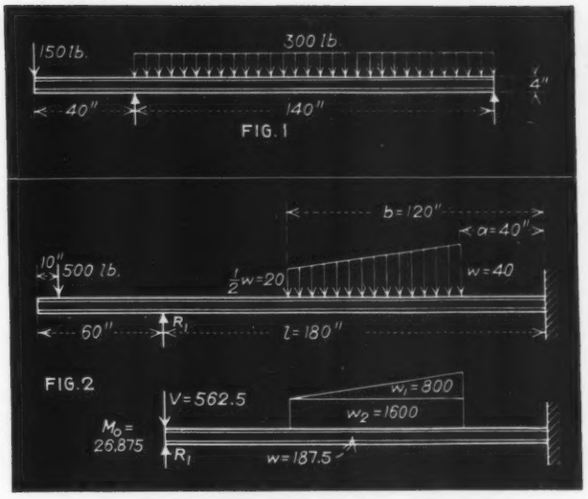


Fig. 1—Calculations for the deflection of this spruce beam at the concentrated 150 lb. load are given in the text to illustrate the use of the tables. Fig. 2—The reaction at the left support of this I-Beam is calculated to illustrate a problem involving a combination of cases

be denoted as W_1 = weight of beam itself equal to 187.5 lb.; W_2 = the uniform part of the trapezoidal load, equal to 1600 lb.; W_3 = the triangular part of the trapezoidal load, equal to 800 lb.; V = the shear imposed by the overhanging portion, equal to 562.5 lb.; M_o = the moment imposed by the overhanging portion, equal to 26,875 in.lb. The reaction R_1 of each of these loads will be separately and independently computed, thus:

For W_1 (Case 23)

$R_1 = (3/8)\ (187.5) = 70.3$ lb.

For W_2 (Case 24)

$$R_1 = (1/8)(1{,}600/180^3)\left\{(4)(180)\ [40^2 + (40)(120) + 120^2] - 40^3 - (40)(120^2) - (40^2)(120) - 120^3\right\} = 425 \text{ lb.}$$

For W_3 (Case 26)

$$R_1 = (1/20)(800/180^3)\left\{[(10)(40)(120) + 15(40^2) + 5(120^2)] \times 180 - 4\ (40^3) - 2\ (40)(120^2) - 3\ (40^2)(120) - 120^3\right\} = 152 \text{ lb.}$$

For M_o (Case 29)

$R_1 = (3/2)\ (26{,}875/180) = 224$ lb.

For V (obvious)

$R_1 = 562.5$ lb.

The actual reaction R_1 because of complete loading is 1,434 lb. the sum of the above.

The deflection at any point of the beam could now be found by regarding the entire beam as a cantilever and calculating the deflections because of its own weight (Case 3), to the triangular part of the trapezoidal load (Case 6), to the uniform part of the trapezoidal load (Case 4), to the 500-lb. load (Case 2) and to an upward load equal to the reaction R_1 (Case 2). The algebraic sum of these deflections would be the deflection caused by the actual loading.

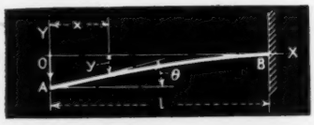
[EDITOR'S NOTE—In addition to the three pages of tabulated formulas given here, additional tabulations will be found on pages 79 and 80, including formulas for statically indeterminate cases. Additional tables will appear in the March number.]

Formulas For Beams

I—STATICALLY DETERMINATE CASES

Loading, Support, Reactions R_1 and R_2, and Vertical Shear V — **Bending Moment M, Max. Bending Moment, Deflection y, Max. Deflection and End Slope θ**

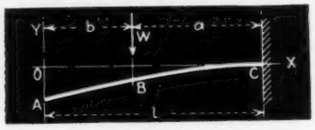
1—CANTILEVER, END LOAD



$M = -Wx$; Max. $M = -Wl$ at B ; $y = -\frac{1}{6}\frac{W}{EI}(x^3 - 3l^2x + 2l^3)$

$R_2 = +W$; $V = -W$; Max. $y = -\frac{1}{3}\frac{Wl^3}{EI}$ at A ; $\theta = +\frac{1}{2}\frac{Wl^2}{EI}$ at A

2—CANTILEVER, INTERMEDIATE LOAD



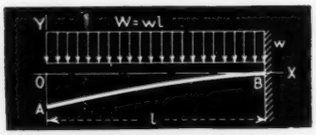
(A to B), $M = 0$; (B to C), $M = -W(x-b)$; Max. $M = -Wa$, at C

(A to B), $y = -\frac{1}{6}\frac{W}{EI}(-a^3 + 3a^2l - 3a^2x)$

(B to C), $y = -\frac{1}{6}\frac{W}{EI}\left[(x-b)^3 - 3a^2(x-b) + 2a^3\right]$

$R_2 = +W$; (A to B), $V = 0$; (B to C), $V = -W$; Max. $y = -\frac{1}{6}\frac{W}{EI}(3a^2l - a^3)$; $\theta = +\frac{1}{2}\frac{Wa^2}{EI}$ (A to B)

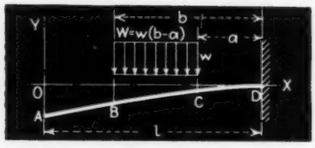
3—CANTILEVER, UNIFORM LOAD



$M = -\frac{1}{2}\frac{W}{l}x^2$; Max. $M = -\frac{1}{2}Wl$, at B ; $y = -\frac{1}{24}\frac{W}{EIl}(x^4 - 4l^3x + 3l^4)$

$R_2 = +W$; $V = -\frac{W}{l}x$; Max. $y = -\frac{1}{8}\frac{Wl^3}{EI}$; $\theta = +\frac{1}{6}\frac{Wl^2}{EI}$, at A

4—CANTILEVER, PARTIAL UNIFORM LOAD



(A to B), $M = 0$; (B to C), $M = -\frac{1}{2}\frac{W}{b-a}(x - l + b)^2$

(C to D), $M = -\frac{1}{2}W(2x - 2l + a + b)$; Max. $M = -\frac{1}{2}W(a+b)$, at D

(A to B), $y = -\frac{1}{24}\frac{W}{EI}\left[4(a^2+ab+b^2)(l-x) - a^3 - ab^2 - a^2b - b^3\right]$

(B to C), $y = -\frac{1}{24}\frac{W}{EI}\left[6(a+b)(l-x)^2 - 4(l-x)^3 + \frac{(l-x-a)^4}{b-a}\right]$

(C to D), $y = -\frac{1}{12}\frac{W}{EI}\left[3(a+b)(l-x)^2 - 2(l-x)^3\right]$

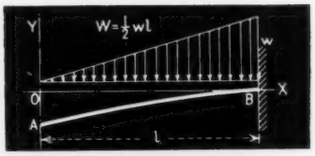
$R_2 = +W$; (A to B), $V = 0$

(B to C), $V = -\frac{W}{b-a}(x - l + b)$

(C to D), $V = -W$

Max. $y = -\frac{1}{24}\frac{W}{EI}\left[4(a^2+ab+b^2)l - a^3 - ab^2 - a^2b - b^3\right]$, at A ; $\theta = +\frac{1}{6}\frac{W}{EI}\left[a^2 + ab + b^2\right]$, (A to B)

5—CANTILEVER, TRIANGULAR LOAD



$M = -\frac{1}{3}\frac{W}{l^2}x^3$; Max. $M = -\frac{1}{3}Wl$, at B ; $y = -\frac{1}{60}\frac{W}{EIl^2}(x^5 - 5l^4x + 4l^5)$

$R_2 = +W$; $V = -\frac{W}{l^2}x^2$; Max. $y = -\frac{1}{15}\frac{Wl^3}{EI}$, at A ; $\theta = +\frac{1}{12}\frac{Wl^2}{EI}$, at A

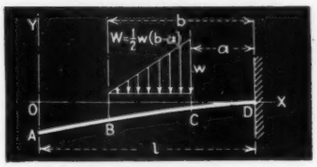
6—CANTILEVER, TRIANGULAR LOAD

$M = -\frac{1}{3}\frac{W}{l^2}\left(3lx^2 - x^3\right)$; Max. $M = -\frac{2}{3}Wl$, at B

$y = -\frac{1}{60}\frac{W}{EIl^2}\left(-x^5 - 15l^4x + 5lx^4 + 11l^5\right)$

$R_2 = +W$; $V = -W\left(\frac{2lx - x^2}{l^2}\right)$; Max. $y = -\frac{11}{60}\frac{Wl^3}{EI}$, at A ; $\theta = +\frac{1}{4}\frac{W}{EI}l^2$, at A

7—CANTILEVER, PARTIAL TRIANGULAR LOAD



(A to B), $M = 0$; (B to C), $M = -\frac{1}{3}\frac{W(x-l+b)^3}{(b-a)^2}$

(C to D), $M = -\frac{1}{3}W(3x-3l+b+2a)$; Max. $M = -\frac{1}{3}W(b+2a)$, at D

(A to B), $y = -\frac{1}{60}\frac{W}{EI}\left[(5b^2+10ba+15a^2)(l-x)-4a^3-2ab^2-3a^2b-b^3\right]$

(B to C), $y = -\frac{1}{60}\frac{W}{EI}\left[(20a+10b)(l-x)^2-10(l-x)^3+5\frac{(l-x-a)^4}{b-a}-\frac{(l-x-a)^5}{(b-a)^2}\right]$

(C to D), $y = -\frac{1}{6}\frac{W}{EI}\left[(2a+b)(l-x)^2-(l-x)^3\right]$

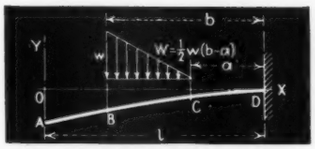
$R_2 = +W$; (A to B), $V = 0$

(B to C), $V = -\frac{W(x-l+b)^2}{(b-a)^2}$

(C to D), $V = -W$

Max. $y = -\frac{1}{60}\frac{W}{EI}\left[(5b^2+10ba+15a^2)l-4a^3-2ab^2-3a^2b-b^3\right]$, at A ; $\theta = +\frac{1}{12}\frac{W}{EI}(3a^2+2ab+b^2)$, (A to B)

8—CANTILEVER, PARTIAL TRIANGULAR LOAD



(A to B), $M = 0$; (B to C), $M = -\frac{1}{3}W\left[\frac{3(x-l+b)^2}{b-a}-\frac{(x-l+b)^3}{(b-a)^2}\right]$

(C to D), $M = -\frac{1}{3}W(-3l+3x+2b+a)$; Max. $M = -\frac{1}{3}W(2b+a)$, at D

(A to B), $y = -\frac{1}{60}\frac{W}{EI}\left[(5a^2+10ab+15b^2)(l-x)-a^3-2a^2b-3ab^2-4b^3\right]$

(B to C), $y = -\frac{1}{60}\frac{W}{EI}\left[\frac{(l-x-a)^5}{(b-a)^2}-10(l-x)^3+(10a+20b)(l-x)^2\right]$

(C to D), $y = -\frac{1}{6}\frac{W}{EI}\left[(a+2b)(l-x)^2-(l-x)^3\right]$

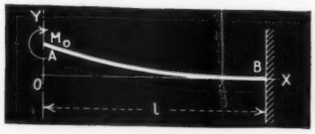
$R_2 = +W$; (A to B), $V = 0$

(B to C), $V = -W\left[1-\frac{(l-a-x)^2}{(b-a)^2}\right]$

(C to D), $V = -W$

Max. $y = -\frac{1}{60}\frac{W}{EI}\left[(5a^2+10ab+15b^2)l-a^3-2a^2b-3ab^2-4b^3\right]$ at A ; $\theta = +\frac{1}{12}\frac{W}{EI}(a^2+2ab+3b^2)$, A to B

9—CANTILEVER, END COUPLE

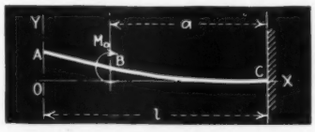


$M = M_0$; Max. $M = M_0$, (A to B)

$y = \frac{1}{2}\frac{M_0}{EI}(l^2-2lx+x^2)$; Max. $y = +\frac{1}{2}\frac{M_0 l^2}{EI}$, at A ; $\theta = -\frac{Ml}{EI}$, at A

$R_2 = 0$; $V = 0$

10—CANTILEVER, INTERMEDIATE COUPLE



(A to B), $M = 0$; (B to C), $M = M_0$; Max. $M = M_0$, (B to C)

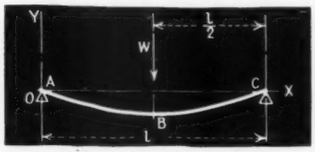
(A to B), $y = \frac{M_0 a}{EI}(l-\frac{1}{2}a-x)$

(B to C), $y = \frac{1}{2}\frac{M_0}{EI}\left[(x-l+a)^2-2a(x-l+a)+a^2\right]$

$R_2 = 0$; $V = 0$

Max. $y = \frac{M_0 a}{EI}(l-\frac{1}{2}a)$, at A ; $\theta = -\frac{M_0 a}{EI}$ A to B

11—END SUPPORTS, CENTER LOAD



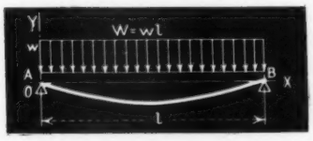
(A to B), $M = +\frac{1}{2}Wx$; (B to C), $M = +\frac{1}{2}W(l-x)$

Max. $M = +\frac{1}{4}Wl$, at B ; (A to B), $y = -\frac{1}{48}\frac{W}{EI}(3l^2x-4x^3)$

Max. $y = -\frac{1}{48}\frac{Wl^3}{EI}$, at B ; $\theta = -\frac{1}{16}\frac{Wl^2}{EI}$, at A ; $\theta = +\frac{1}{16}\frac{Wl^2}{EI}$, at C

$R_1 = +\frac{1}{2}W$; $R_2 = +\frac{1}{2}W$; (A to B), $V = +\frac{1}{2}W$; (B to C), $V = -\frac{1}{2}W$

12—END SUPPORTS, UNIFORM LOAD



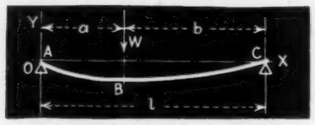
$M = \frac{1}{2}W\left(x-\frac{x^2}{l}\right)$; Max. $M = +\frac{1}{8}Wl$, at $x = \frac{1}{2}l$

$y = -\frac{1}{24}\frac{Wx}{EIl}\left(l^3-2lx^2+x^3\right)$; Max. $y = -\frac{5}{384}\frac{Wl^3}{EI}$ at $x = \frac{l}{2}$

$R_1 = +\frac{1}{2}W$; $R_2 = +\frac{1}{2}W$; $V = \frac{1}{2}W\left(1-\frac{2x}{l}\right)$

$\theta = -\frac{1}{24}\frac{Wl^2}{EI}$, at A ; $\theta = +\frac{1}{24}\frac{Wl^2}{EI}$ at B

13—END SUPPORTS, INTERMEDIATE LOAD



(A to B), $M = +W\frac{b}{l}x$; (B to C), $M = +W\frac{a}{l}(l-x)$; Max. $M = +W\frac{ab}{l}$, at B

(A to B), $y = -\frac{Wbx}{6EIl}\left[2l(l-x) - b^2 - (l-x)^2\right]$

(B to C), $y = -\frac{Wa(l-x)}{6EIl}\left[2lb - b^2 - (l-x)^2\right]$

$R_1 = +W\frac{b}{l}$; $R_2 = +W\frac{a}{l}$

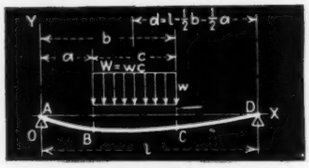
(A to B), $V = +W\frac{b}{l}$

(B to C), $V = -W\frac{a}{l}$

Max. $y = -\frac{W\,a\,b}{27\,E\,I\,l}(a+2b)\sqrt{3a(a+2b)}$, at $x = \frac{1}{3}\sqrt{a(a+2b)}$ when $a > b$

$\theta = -\frac{1}{6}\frac{W}{EI}\left(bl - \frac{b^3}{l}\right)$ at A ; $\theta = +\frac{1}{6}\frac{W}{EI}\left(2bl + \frac{b^3}{l} - 3b^2\right)$, at B

14—END SUPPORTS, PARTIAL UNIFORM LOAD



(A to B), $M = R_1x$; (B to C), $M = R_1x - W\frac{(x-a)^2}{2c}$

(C to D), $M = R_1x - W\left(x - \frac{1}{2}a - \frac{1}{2}b\right)$; Max. $M = W\frac{d}{l}\left(a + \frac{cd}{2l}\right)$, at $x = a + \frac{cd}{l}$

(A to B), $y = \frac{1}{48EI}\left\{8R_1(x^3 - l^2x) + Wx\left[\frac{8d^3}{l} - \frac{2bc^2}{l} + \frac{c^3}{l} + 2c^2\right]\right\}$

(B to C), $y = \frac{1}{48EI}\left\{8R_1(x^3 - l^2x) + Wx\left[\frac{8d^3}{l} - \frac{2bc^2}{l} + \frac{c^3}{l} + 2c^2\right] - 2W\frac{(x-a)^4}{c}\right\}$

(C to D), $y = \frac{1}{48EI}\left\{8R_1(l^3 - l^2x) + Wx\left[\frac{8d^3}{l} - \frac{2bc^2}{l} + \frac{c^3}{l}\right] - 8W\left(x - \frac{1}{2}a - \frac{1}{2}b\right)^3 + W\left[2bc^2 - c^3\right]\right\}$

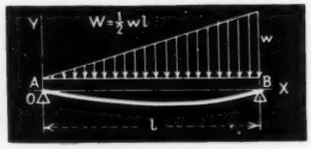
$R_1 = W\frac{d}{l}$ $R_2 = \frac{W}{l}$; $\left(\frac{1}{2}a + \frac{1}{2}b\right)$

(A to B), $V = R_1$

(B to C), $V = R_1 - W\frac{x-a}{c}$; (C to D), $V = R_1 - W$

$\theta = \frac{1}{48EI}\left[-8R_1l^2 + W\left(\frac{8d^3}{l} - \frac{2bc^2}{l} + \frac{c^3}{l} + 2c^2\right)\right]$ at A ; $\theta = \frac{1}{48EI}\left[16R_1l^2 - W\left(24d^2 - \frac{8d^3}{l} + \frac{2bc^2}{l} - \frac{c^3}{l}\right)\right]$ at B

15—END SUPPORTS, TRIANGULAR LOAD



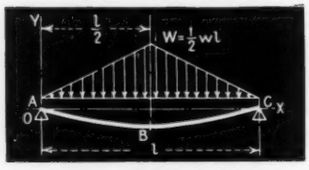
$M = \frac{1}{3}W\left(x - \frac{x^3}{l^2}\right)$; Max. $M = 0.128\,Wl$, at $x = l\left(\frac{\sqrt{3}}{3}\right) = 0.5774\,l$

$y = -\frac{1}{180}\frac{Wx}{EIl^2}(3x^4 - 10l^2x^2 + 7l^4)$; Max. $y = -0.01304\frac{Wl^3}{EI}$, at $x = 0.519\,l$

$\theta = -\frac{7}{180}\frac{Wl^2}{EI}$, at A ; $\theta = +\frac{8}{180}\frac{Wl^2}{EI}$, at B

$R_1 = \frac{1}{3}W$; $R_2 = \frac{2}{3}W$; $V = W\left(\frac{1}{3} - \frac{x^2}{l^2}\right)$

16—END SUPPORTS, TRIANGULAR LOAD



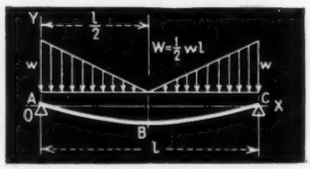
(A to B), $M = \frac{1}{6}W\left(3x - 4\frac{x^3}{l^2}\right)$; (B to C), $M = \frac{1}{6}W\left[3(l-x) - 4\frac{(l-x)^3}{l^2}\right]$

Max. $M = \frac{1}{6}Wl$, at B ; (A to B), $y = \frac{1}{6}\frac{Wx}{EIl^2}\left(\frac{1}{2}l^2x^2 - \frac{1}{5}x^4 - \frac{5}{16}l^4\right)$

Max. $y = -\frac{1}{60}\frac{Wl^3}{EI}$, at B ; $\theta = -\frac{5}{96}\frac{Wl^2}{EI}$, at A ; $\theta = +\frac{5}{96}\frac{Wl^2}{EI}$, at C

$R_1 = \frac{1}{2}W$; $R_2 = \frac{1}{2}W$; (A to B), $V = \frac{1}{2}W\left(1 - 4\frac{x^2}{l^2}\right)$; (B to C), $V = -\frac{1}{2}W\left(1 - 4\frac{(l-x)^2}{l^2}\right)$

17—END SUPPORTS, TRIANGULAR LOAD



(A to B), $M = \frac{1}{2}W\left(x - 2\frac{x^2}{l} + \frac{4}{3}\frac{x^3}{l^2}\right)$

(B to C), $M = \frac{1}{2}W\left[(l-x) - 2\frac{(l-x)^2}{l} + \frac{4}{3}\frac{(l-x)^3}{l^2}\right]$; Max. $M = \frac{1}{12}Wl$, at B

(A to B), $y = \frac{1}{12}\frac{W}{EI}\left(x^3 - \frac{x^4}{l} + \frac{2}{5}\frac{x^5}{l^2} - \frac{3}{8}l^2x\right)$; Max. $y = -\frac{3}{320}\frac{Wl^3}{EI}$, at B

$\theta = -\frac{1}{32}\frac{Wl^2}{EI}$, at A ; $\theta = +\frac{1}{32}\frac{Wl^2}{EI}$, at B

$R_1 = \frac{1}{2}W$; $R_2 = \frac{1}{2}W$

(A to B), $V = \frac{1}{2}W\left(\frac{l-2x}{l}\right)^2$; (B to C), $V = -\frac{1}{2}W\left(\frac{2x-l}{l}\right)^2$

(Continued on page 79)

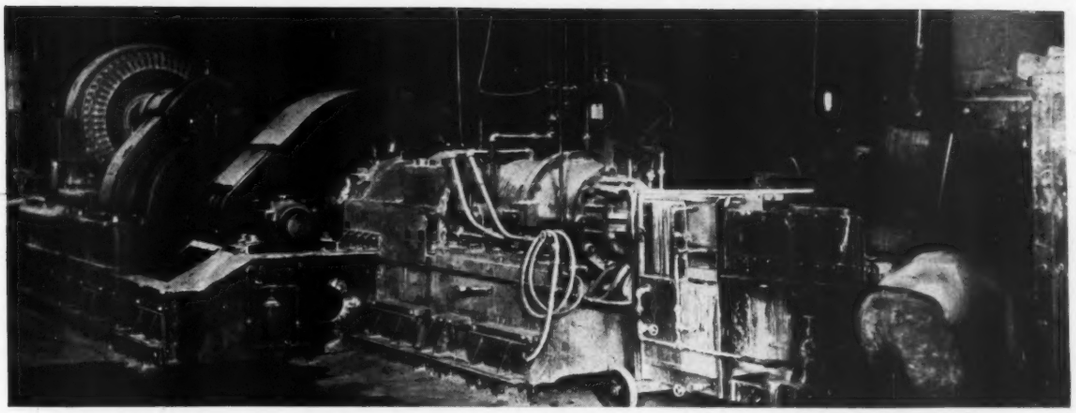


Fig. 1—The 15-in. Gordon plasticator weighing nearly 30 tons with its 250 hp. synchronous motor and reduction gear all on one base mounted on flat type rubber mountings

Rubber Mountings

for Large Industrial Machines—I

STUART H. HAHN
Physical Research Laboratory, The B. F. Goodrich Company

The demand for vibration isolation has reached the field of heavy machinery. Three successful installations described here show how such machines can be designed originally with rubber mounting systems

THE problem of noise and vibration has become so acute in modern life that relief is being demanded by the public for its own comfort and well-being (Reference 1) and by the technical world because of the destructive effects of vibrations on buildings and machinery. Vibration is uneconomical.

Reduction of vibration at its source by improved design and construction has been the trend for years but the increasing demand for greater output and higher speeds has largely nullified the effects of these improvements. Thus has come about the need for cushioning residual vibration and, as in the automotive industry, we find rubber mountings used to an increasing degree on many types of light machinery. Although the use of compression-type rubber mountings under fairly heavy power generating equipment has been discussed in Germany (Reference 2), it is believed that no applications of rubber mountings of any type to heavy industrial machinery have been reported in this country.

Shear-type rubber mountings have been effectively isolating the vibrations of five large machines in the main plant of The B. F. Goodrich Company for periods of from one to more than three years. The total weight of these units is about 430,000 lb. All are installed on the fourth or higher floors of buildings of various types used for manufacturing, office, or machine-shop purposes. Experience with these installations has been uniformly favorable. The degree of cushioning obtained, and the durability and stability of the mountings has been entirely satisfactory. At the present time four more machines totaling in weight about 200,000 lb. are being installed on rubber mountings. The application of rubber mountings to large industrial machines has emerged from the experimental stage.

Mounting for a 15-In. Gordon Plasticator

The oldest of the installations mentioned employs flat plate, single shear-type mountings to support a 15-inch Gordon plasticator, a product of Farrel-Birmingham Company, as shown in Fig. 1. This machine with its 250 hp. synchronous motor and reduction gear weighs nearly 30 tons. Resembling other screw-type extrusion machines, it is used to soften or "plasticate" crude rubber which is fed into its hopper in 37-lb. pieces and leaves the die of the machine in the form of a continuous sheet of plasticized rubber.

Rubber mountings and a welded steel sub-base were designed to fit a unit originally intended to be grouted and bolted to its foundation. Favorable elements included the adaptability of the machine to a unit as-

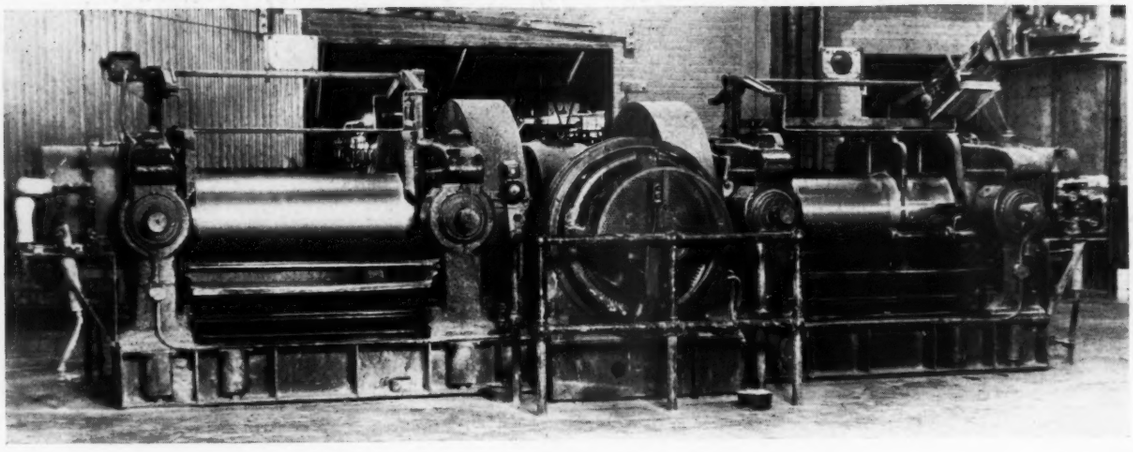


Fig. 3—Double unit 60-in. rubber warming mill. The cylindrical pieces in the sub-base are split shell tubular rubber mountings

sembly and ample head room allowance. The heavy torques and internal stresses involved and the lack of a rigid, unitary sub-base were unfavorable to the design. A welded base was made principally of 20-in. I-beam members. It was bolted in a massive joint to the rear of the Meehanite bed casting of the plasticator. The low speed pinion shaft driving the plasticator bull gear carries peak loads exceeding 250 hp. at 90 r. p. m. Provision of a rigid support for the heavily loaded main drive pinion was aided by the adoption of a short continuous shaft with three bearing supports for the main pinion and the gear of the speed reduction unit in place of the original longer and coupled two-piece shaft with four bearings.

The use of existing equipment dictated the relatively low motor speed of 514.3 r.p.m. Possible dynamic unbalance of the motor and the attached parts would produce the lowest frequency of any vibrating force likely to cause trouble. In order to isolate this lowest disturbing frequency satisfactorily, the ideal mounting design would therefore, have required a sufficient rubber thickness to reduce the natural or resonant frequencies of the machine to about 2 cycles per second (Reference 3). Considering only vibration in the vertical plane, isolation of unbalanced vibration from the motor would have required that the rubber mountings have a static deflection of about $2\frac{3}{8}$ in. This could not have been obtained economically in the flat or sandwich-type of mounting. It was therefore decided to design the mountings so that they would give the machine a resonant frequency sufficiently low to isolate the lowest disturbing frequency from the gears (about 90 cycles per sec. from the bull gear and its pinion) yet high enough to avoid more than very slight amplification of any disturbance resulting from motor unbalance. Dynamic unbalance of the motor was corrected satisfactorily by an experienced mechanic without dismantling the machine.

Fig. 2 — Large sandwich or flat plate mountings with a load capacity of 4,600 lb., used on the 15-in. Gordon plasticator

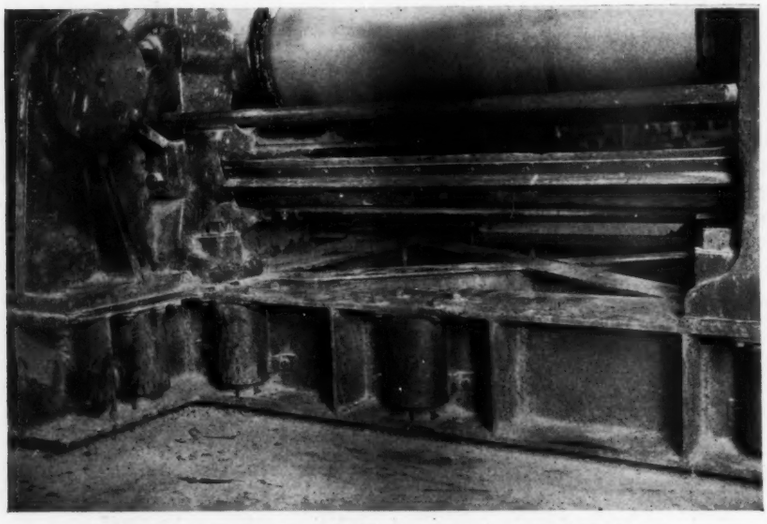


Fig. 4—Close-up of sub-base of the mill showing how the tubular rubber mountings were applied

Rubber mountings of the sandwich-type were designed to give the machine a fundamental resonant frequency of about 16 cycles per sec. in the vertical translatory mode of vibration. Each has a load capacity of about 4,600 lb. (unit shearing stress 20 lb. per sq. in.) and has at that load a spring constant of about 90,000 lb. per in. One of these mountings is shown in Fig. 2. The main members consist of one-half inch

steel plates to which the rubber is vulcanized. Adhesion between rubber and metal was secured by brass-plating the metal parts and vulcanizing each unit under pressure in a mold. These mountings will carry continuous loads well in excess of 200 per cent of the rated load.

Warm-up Mills

A typical 60-inch rubber mounted double mill unit is shown in Fig. 3. The oldest of these units has been in continuous service for nearly three years. As shown in the sub-base in Fig. 4, these units are supported by tubular rubber shear mountings, Fig. 5, which are bolted to the outer side of the upper flange of the wide flange beam used in the sub-base (Reference 4). This is the first application of large tubular mountings to industrial machinery of which we have knowledge.

The design of the mill-unit sub-base was relatively less difficult than that of the Gordon plasticator because of the lower torques and gear tooth loads involved. Its compact and symmetrical arrangement also aided in simplifying the design. The peak power loads delivered by the main pinions rotating at about 120 r.p.m. are of the order of 80 horsepower.

The design of the mill mountings was dictated by the low value of the minimum disturbing frequency, that of the tooth impacts which may be set up by the roll neck connecting gears. This frequency was raised to a reasonable value of about 10 cycles per sec. by the use of staggered tooth gears. The fundamental frequencies of the various critical modes of vibration of these mill units are all 3 to 4 cycles per sec. The static deflection of the mountings when carrying the load of the complete assembly is about 1 in. In mountings of the dimensions used this value may be varied over a range of about plus or minus 30 per cent by the use of different rubber compounds.

These tubular mountings are made up by vulcanizing rubber of suitable grade to a central seamless steel tube and to semi-cylindrical outer metal shells separated by suitable gaps. The unit is molded to an outside diameter somewhat greater than the inside diameter of the housing tube so that the rubber is compressed when the unit is pressed into its housing. The degree of compression is so chosen that shrinkage stresses set up in the rubber during vulcanization are relieved. This factor, together with the use of tapered ends, permits the use of much higher maximum unit shearing stresses between rubber and metal than would be permissible without this construction. These units have a load capacity of 4,000 to 5,300 lb. each and a spring constant of from 3,800 to 7,000 lb. per in. of deflection, depending on the type of rubber compound.

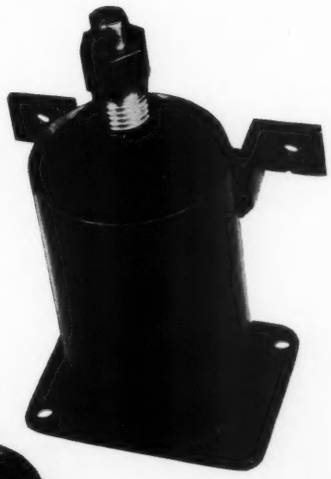


Fig. 5—Split shell tubular mountings, having a load capacity of 4,000 to 5,300 lb. and a spring constant of from 3,800 to 7,000 lb. per in. used on the 60-in. warming mill

20-*In. Gordon Plasticator*

Convinced that the experimental state in the use of rubber mountings for heavy machinery had been passed, the Goodrich engineering staff undertook the design of a sub-base and rubber mounting arrangement for a new Gordon plasticator, shown in Fig. 6, the largest machine of its kind. The design presented a number of serious problems, among them the following:

1. The large torque involved in delivering peaks of about 700 hp. at 123 r.p.m. to the plasticator pinion necessitated the use of a unitary assembly. The torque delivered by the low speed shaft would be almost sufficient to overturn the reduction gear and motor section of the machine if it were mounted separately from the plasticator proper. This comparatively large transfer of load on the mountings would have created such irregular loading that trouble with variable creep in the rubber and consequent coupling misalignment would have been difficult to avoid. Possible instability in the transverse rocking mode of vibration would also have required careful study.

2. The necessity for maintaining rigid support for the shaft carrying the wide (24-inch) main drive herringbone pinion called for an unusually rigid sub-base construction which would

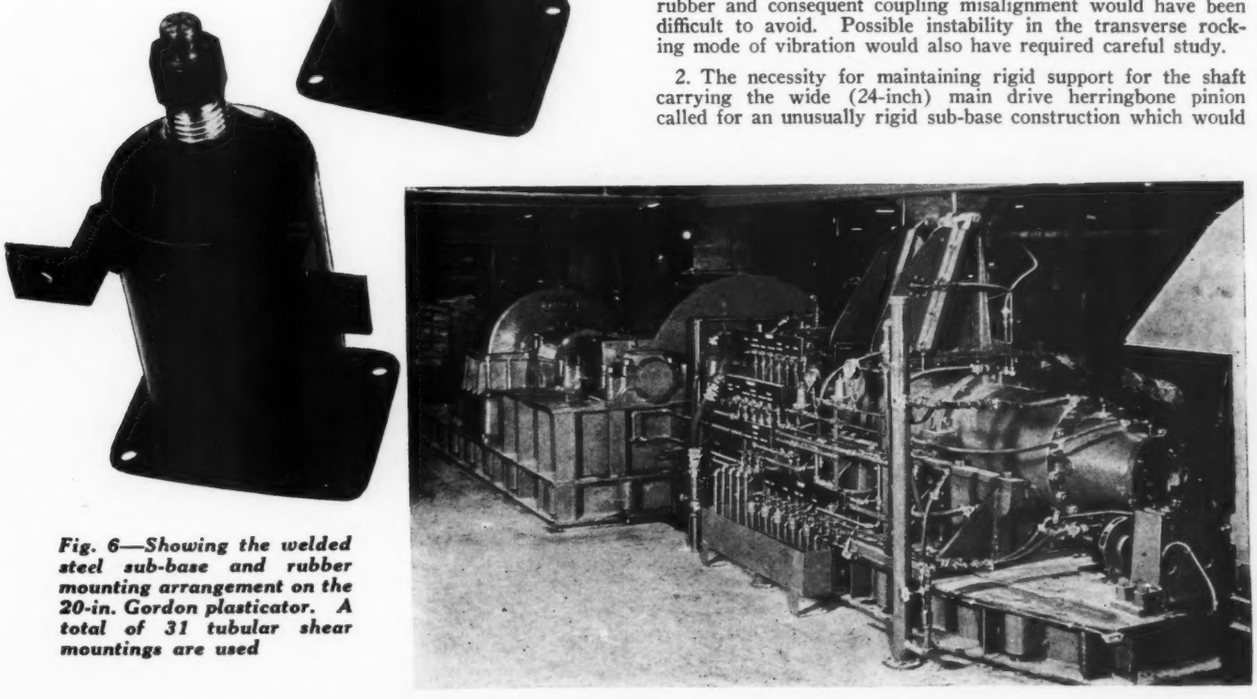


Fig. 6—Showing the welded steel sub-base and rubber mounting arrangement on the 20-in. Gordon plasticator. A total of 31 tubular shear mountings are used

have been difficult to secure in a multiple unit design. This gear is subject to tooth loads on the order of 1,500 lb. per in. of width at a pitch line speed of about 720 ft. per min. The pinion shaft bearings produce concentrated loads on the sub-base of about 20,000 lb. each.

3. A large eccentric bending load on the bed casting produced by the forward thrust of the rubber passing through the cylinder of the machine, held in place on longitudinal ways by an adjusting screw about 18 in. below the centerline of the rotor, together with the reaction on the rotor thrust bearing, causes an estimated bending moment of about 500,000 in. lb. The large radial roller bearings and Kingsbury thrust bearing require particularly rigid support.

4. Head room restrictions would not permit raising the machine more than 6 in. above the floor.

The welded steel sub-base, shown in Fig. 7, is 42 ft. long and 11 ft. wide, and weighs about 22,000 lb. It was designed to carry the complete assembly as a unit. The original design of the plasticator bed casting was modified by the manufacturers to increase its stiffness and adapt it to mounting on the steel sub-base. To the base were attached 31 tubular shear mountings of the type used so successfully in mounting several double mill units like the one which has been described. These mountings were so located as to obtain uniform mounting loads together with minimum bending moments in the sub-base. The ruling factor in the choice of mounting design was the frequency of the slowest rotating part likely to be a source of vibration. This frequency, 12 cycles per sec. originating in the 720 r.p.m. synchronous motor and connected parts, is effectively isolated by mountings having a natural frequency of vertical oscillation under rated load of slightly more than 3 cycles per second.

REFERENCES—PART I

1—Series of papers by S. A. Moss, Johns-Hopkins University, in *S.A.E. Journal*, starting September 1929.

2—"Gummifundamente," K. Oeser, Kautschuk 10, 8, 121-126, August 1934.

3—S. Timoshenko, "Vibration Problems in Engineering," (Van Nostrand) or any standard text on vibration problems.

4—U.S. Pat. 1,958,141 (F. L. Haushalter to B. F. Goodrich Company, May 8, 1934).

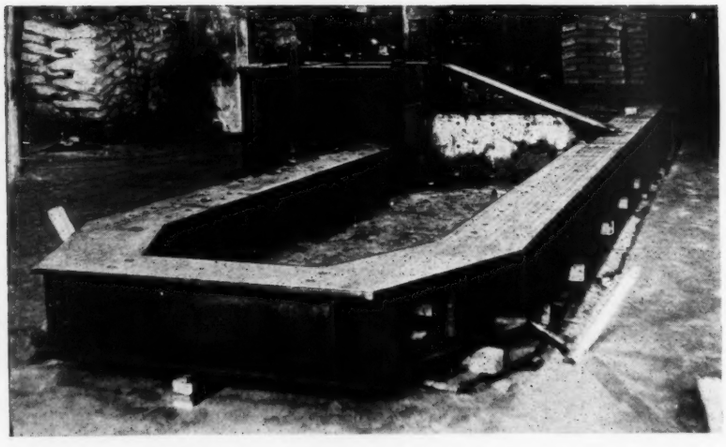


Fig. 7—Welded steel sub-base, 42 ft. long and 11 ft. wide and weighing about 22,000 lb., designed to carry the complete assembly of the 20-in. Gordon plasticator

Notwithstanding the relatively wide variation in concentration of loads in various parts of the sub-base, mountings of uniform size and construction were used successfully to support and cushion the load uniformly. Distribution of the mountings was based on careful estimates of the weight distribution in the various parts of the machine, which were checked, wherever possible, by actual weighing. An estimate of the polar moment of inertia of the machine was made for the purpose of calculating the probable resonance frequency of the transverse rocking mode of vibration. This frequency was found to be safely below that of the possible transverse disturbance which might be set up by unbalance in the motor, the shaft of which was located at a considerable distance above the level of the mountings.

In general, any body mounted on a system of elastic supports which are not in themselves constrained to any definite "deflection path" will have six degrees of freedom. Three of these are the components of its translatory or "straight-line" motions which are usually referred to three mutually perpendicular axes, one of which is vertical. The other three are the rotational components of motion likewise usually referred to three mutually perpendicular axes. Considering the more usual types of industrial machines, it is often true that only one of these six degrees of freedom — that of vertical translation—has much importance. Vertical oscillatory forces are the more usual sources of annoying vibration.

However, it may sometimes be necessary to make an analysis of the dynamics of a spring-mounted machine in several of its degrees of freedom, especially if there are rotating unbalanced forces, or reciprocating forces in horizontal planes. Such forces may frequently be found to act transversely to the longitudinal axis of the machine so that transverse translatory or rocking motions will be set up either alone or in combination. Hence, the natural frequency of the machine in one or both of these (and possibly in other) modes of vibrations must be studied. To calculate the natural frequency of rocking, the flywheel effect or polar moment of inertia of the machine and the angular spring constant or torque per unit of angular deflection set up by the springs at their known locations must be determined. Such angular spring constants will usually depend on a combination of the vertical and horizontal deflection characteristics of the individual spring units. In general, the horizontal characteristic of any spring intended primarily for resisting vertically applied force will also depend partly on the elastic characteristics of its attachment to the foundation.

The success of these rubber mounting installations is proved by the fact that almost all new machinery installations in the upper floors of buildings of the Akron plants of the Goodrich Company which have been made in the past four years have been equipped with rubber mountings. When present work is completed eleven units with a gross weight of about 675,000 lb. will be in service. The reduction in building vibration secured through the use of the mountings has been marked in every installation. In a few cases records of vibration with and without rubber mountings have been secured with vibrographs. Some of these records will be shown in Part II of this article.

Principles of Design for NON-CIRCULAR GEARS

PART III — Elliptical Rolling Curves

Equations for external and internal mutually rolling curves having one or many lobes generated by means of an ellipse rotating about one focus

J. J. PESQUEIRA

ELLIPTICAL rolling curves, as the name suggests are derived from the general equations of an ellipse which has one focus at the center of rotation.

Beginning with the ellipse, as shown in Fig. 9, taking focus F as the origin of coordinates, the half-line FA, which is coincident with the major axis, as the principal ray, p as the semi-latus rectum and e as the eccentricity, the polar equation is

$$r = \frac{p}{(1 - e\cos\theta)} \tag{36}$$

With $2a$ as the length of the major axis, $2b$ as the length of the minor axis, and $2c$ as the focal distance, the following relations hold

$$e = \frac{c}{a} = \sqrt{1 - \left(\frac{b}{a}\right)^2}$$

$$p = a(1 - e^2) = \frac{b^2}{a}$$

$$b = a\sqrt{1 - e^2}$$

The maximum radius vector FA is

$$r_{max} = \frac{p}{1 - e}$$

and the minimum radius vector FA' is

$$r_{min} = \frac{p}{1 + e}$$

But when the other focus F' is used as origin, and the half-line $F'A$ as the principal ray, the polar equation is

$$r' = \frac{p}{(1 + e\cos\theta')} \tag{37}$$

By definition of the ellipse

$$r + r' = 2a \tag{38}$$

Let L, shown in Fig. 10, be an ellipse and a rolling curve the center of rotation of which is O at one focus; and let L' be the derived rolling curve, having center of rotation at O'. Let $OO' = q$. The polar equation of L', in parametric form, will then be, by Equations (9) and (10)

$$r' = q - \frac{p}{(1 - e\cos\theta)} \tag{39}$$

$$\theta' = \int_0^{\theta} \left\{ \frac{n/(1 - e\cos\theta)}{q - [p/(1 - e\cos\theta)]} \right\} d\vartheta = p\int_0^{\theta} \frac{d\theta}{(q - p) - qe\cos\theta}$$

$$= \frac{2p}{\sqrt{(q - p)^2 - q^2e^2}} \tan^{-1}\left[\frac{\sqrt{(q - p)^2 - q^2e^2}}{(q - p) - qe} \tan\tfrac{1}{2}\theta \right] \tag{40}$$

where $(q - p)^2 > q^2e^2$. The rolling curve L', as given by Equations (39) and (40) will in general be open.

Let L' be closed and have n' lobes. Equation (14) will give here, for $n' = n'$, and $n = 1$, L being single-lobed,

$$\frac{2\pi}{n'} = \frac{2\pi p}{\sqrt{(q - p)^2 - q^2e^2}}$$

from which

$$q = \frac{p}{1 - e^2}\left[1 \pm \sqrt{1 + (n'^2 - 1)(1 - e^2)} \right] \tag{41}$$

Substituting this value of q in Equation (40)

$$\left[\pm \sqrt{1 + (n'^2 - 1)(1 - e^2)} - e \right] \tan\tfrac{1}{2}(n'\theta')$$

$$= (1 + e)\tan\tfrac{1}{2}\theta \tag{42}$$

Putting this result in the form

$$R^2 \tan^2\tfrac{1}{2}(n'\theta') = \tan^2\tfrac{1}{2}\theta$$

where

$$R = \frac{\pm\sqrt{1 + (n'^2 -)(1 - e^2)} - e}{1 + e}$$

and introducing the trigonometric identity

$$\tan^2\tfrac{1}{2}\phi = \frac{1 - \cos\phi}{1 + \cos\phi}$$

Equation (42) becomes

$$\cos\theta = \frac{1 - R^2 + (1 + R^2)\cos(n'\theta')}{1 + R^2 + (1 - R^2)\cos(n'\theta')} \tag{43}$$

Substituting the value of q as given by Equation (41) and that of $\cos\theta$ as given by Equation (43), in the parametric Equation (39), this becomes

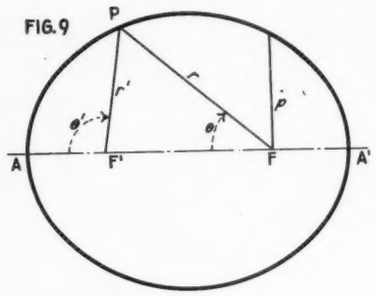
$$r' = \frac{n'^2 p}{\pm\sqrt{1 + (n'^2 - 1)(1 - e^2)} + e\cos(n'\theta')} \tag{44}$$

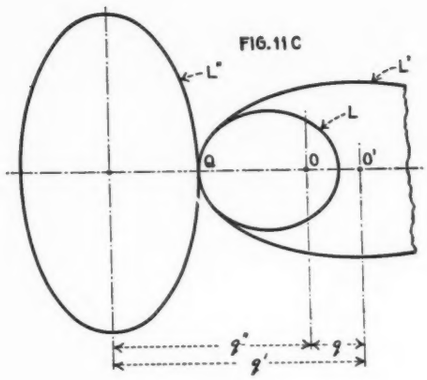
which is the equation of L' in polar form.

In Equation (44) the radical is always greater in absolute value than e, therefore, when the plus sign before the radical is used, the equation becomes

$$r' = \frac{n'^2 p}{\sqrt{1 + (n'^2 - 1)(1 - e^2)} + e\cos(n'\theta')} \tag{44'}$$

in which r' is seen to remain always positive. Since r, in L, as given by Equation (36) also remains positive, r and r' being thus always similarly signed, the curves L and L' as given by Equations (36) and (44'),





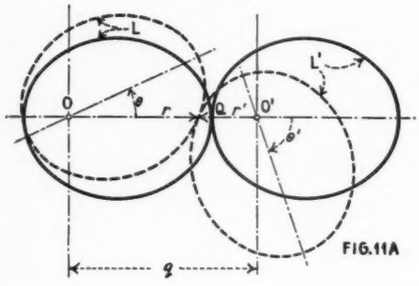


FIG. 11A

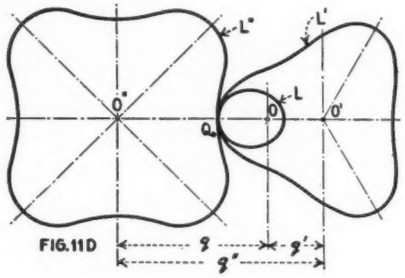
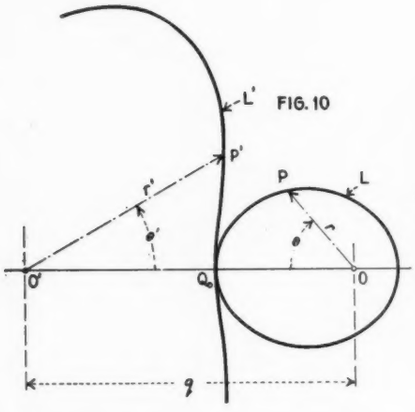


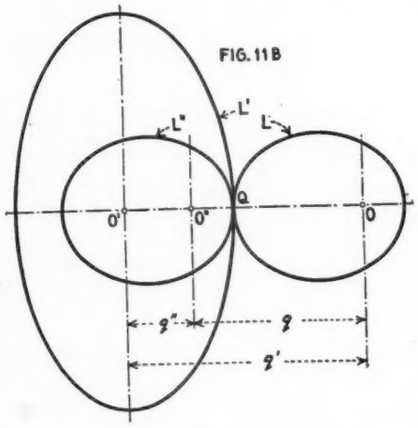
Fig. 9—The simple ellipse is the basis from which an important group of rolling curves is derived

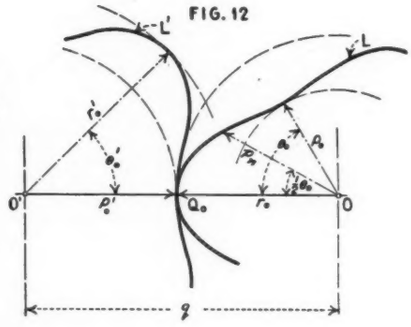
Fig. 10—An ellipse rotating about one focus is used to develop a mutual rolling curve with enter of rotation at O'

Fig. 11—A few particular cases of elliptical rolling curves, (A) Derived curve having one lobe, (B) generated curve L' has two lobes and rotates about its center, (C) External and internal rolling curves having two lobes generated by ellipse, (D) Internal rolling curve of three lobes and an external curve of four lobes with generating ellipse

Fig. 12—Linear elements of a lobed curve used to derive rolling curve equations in terms of maximum and minimum radii vectors







respectively, are in external engagement. But, if the minus sign before the radical in Equation (44) is retained, it becomes

$$r' = -\frac{n'^2\, p}{\sqrt{1 + (n'^2 - 1)(1 - e^2)} - e\cos(n'\,\theta')} \qquad (44'')$$

in which r' is seen to remain always negative, r and r' are in this case oppositely signed, and the curves L and L' will, therefore, roll with each other in internal engagement. The curves represented by (44′) and (44″) are clearly identical in shape, differing only in their position with reference to the frame of coordinates used. Angularly, the two curves are displaced from each other by the amount π/n', or half a period (half a lobe); that is to say, they are out of phase by that amount.

The curves of Equations (44′) and (44″), when rolling with the curve of Equation (36) in external or internal engagement, respectively, will, of course, rotate on different centers. The center distance q, in each case, is found from Equation (41), where the sign before the radical is selected to correspond to the choice of sign in Equation (44). Here q is seen to be positive or negative, depending upon whether the engagement of L and L' is external or internal.

It has been shown that a closed elliptical curve L' of n' lobes may be generated by the ellipse with center of rotation at one focus, either in external or internal engagement. The aplication of the Theorem of Interchangeability leads to the following statement:

Two closed elliptical rolling curves, each having any number of lobes, both being derived from the same ellipse with a focus as center of rotation, are mutually rolling curves, either externally or internally. Since each one of these rolling curves can be considered as derived from the other, and since they are any two curves, the statement can be interpreted to mean that all rolling curves derived from the same elliptical curve (not necessarily the ellipse itself) are mutually rolling curves, either externally or internally.

A few particular cases of elliptical rolling curves will be reviewed.

Case (A) where $n' = 1$. Equations (41) and (44) become for $n' = 1$

$$q = \frac{p}{1 - e}(1 \pm 1) = a(1 \pm 1)$$

$$r' = \frac{p}{\pm 1 + e\cos\theta'}$$

If the plus sign is retained in these equations, then $q = 2a$, and the second equation is seen to be that of an ellipse identical in shape to the generating ellipse. This pair of rolling ellipses is shown in Fig. 11A,

where the solid lines represent the ellipses in the original positions, and the dotted outline represents the same curves after having been displaced angularly in their rolling action.

The choice of the minus sign would give $q = 0$, and $r' < 0$, and the curves will coincide with each other, turning on the same center of rotation. This is an extreme case of internal engagement.

Case (B) where $n' = 2$. The substitution of the value $n' = 2$ in Equations (41) and (44) gives

$$q = a\,(1 \pm \sqrt{4 - 3\,e^2})$$

$$r' = \frac{4p}{\pm\sqrt{4 - 3e^2} + e\cos 2\theta'}$$

where $a = p/(1 - e)$. The second of these equations is that of the ellipse L', shown in Fig. 11B, the center of rotation of which is at the center of figure. L' is also shown there as engaging the generating ellipse L externally. In the same figure, another ellipse, L'', identical to L, is shown engaging L externally and L' internally.

Case (C) where $n' = 2$, and $n'' = 2$. Referring to Fig. 11C, L' is the ellipse

$$r' = -\frac{4p}{\sqrt{4 - 3e^2} - e\cos 2\theta'}$$

with center of rotation O' at the center of figure, generated internally by the ellipse L with center of rotation O at one of its foci. The curve L'' is the ellipse

$$r'' = \frac{4p}{\sqrt{4 - 3e^2} + e\cos 2\theta''}$$

with the center of rotation O'' at the center of figure, generated externally by L. With reference to the notation in the figure, the distance between the centers of rotation of L' and L'' is

$$q' = q + q'' = a\,(1 + \sqrt{4 - 3e^2}) + [-a\,(1 - \sqrt{4 - 3\,e^2})] = 2a\sqrt{4 - 3e^2}$$

Case (D) where $n' = 3$, and $n'' = 4$. Fig. 11D shows a three-lobed curve L' in external engagement with the four-lobed curve L''. The generating ellipse L is in internal engagement with L' and in external engagement with L''. The distance between the centers of rotation of L' and L'' is,

$$q'' = q + q'$$

By Equation (41)

$$q = a\,(1 + \sqrt{16 - 15\,e^2})$$

$$q' = a\,(1 - \sqrt{9 - 8e^2})$$

A polar equation for an n-lobed elliptical curve may be obtained without the explicit presence there of the constants e and p belonging to the one-lobed curve or generating ellipse. That equation may be derived in terms of certain linear elements of the n-lobed curve itself. Such an equation will be derived in terms of the maximum and minimum radii vectors.

The polar equation of an n-lobed curve is

$$r = \frac{n^2\,p}{\sqrt{1 + (n^2 - 1)\,(1 - e^2)} - e\cos n\theta}$$

which can be written

$$r = \frac{p_n}{1 - e_n\cos n\,\theta} \qquad (s)$$

where

$$p_n = \frac{n^2 p}{\sqrt{1 + (n^2 - 1)\,(1 - e^2)}} \text{ and } e_n = \frac{e}{\sqrt{1 + (n^2 - 1)\,(1 - e^2)}}$$

Referring to Fig. 12, let r_o be the maximum radius vector of the n-lobed curve L, and let ρ_o be the minimum radius vector of the same curve. The angle between adjacent maximum and minimum radii vectors is $\theta_o = \pi/n$, or half the period of the curve. Putting successively $\theta = 0$, and $\theta = \theta_o$ in Equation (s), the equation becomes

$$r_o = \frac{p_n}{1 - e_n}, \text{ and } \rho_o = \frac{p_n}{1 + e_n}$$

whence

$$p_n = \frac{2r_o\ \rho_o}{r_o + \rho_o}, \text{ and } e_n = \frac{r_o - \rho_o}{r_o + p_o}$$

The substitution of these values in (s) gives

$$r = \frac{2r_o\ \rho_o}{r_o + \rho_o - (r_o - \rho_o)\cos(n\theta)} \qquad (45)$$

which is the equation sought.

Similarly, for the derived curve L' can be obtained

$$r' = \frac{p_{n'}}{\pm 1 + e_{n'}\cos(n'\,\theta')} \qquad (T)$$

where

$$p_{n'} = \frac{n'^2\,p}{\sqrt{1 + (n'^2 - 1)\,(1 - e^2)}}, \text{ and } e_{n'} = \frac{e}{\sqrt{1 + (n'^2 - 1)\,(1 - e^2)}}$$

and also

$$r' = \frac{2r_{o'}\ \rho_{o'}}{\pm(r_{o'} + \rho_{o'}) + (r_{o'} - \rho_{o'})\cos(n'\,\theta')} \qquad (46)$$

where $r_{o'}$ and $p_{o'}$ are the maximum and minimum radii vectors, respectively, of the derived n'-lobed curve L'.

The distance between the centers of rotation of L and L' is clearly

$$q = r_o + \rho_{o'} = r_{o'} + \rho_o \qquad (U)$$

Now, since

$$\frac{p_n}{e_n} = \frac{2r_o\ \rho_o}{r_o - \rho_o} = \frac{n^2\,p}{e}$$

$$\frac{p_{n'}}{e_{n'}} = \frac{2r_{o'}\ \rho_{o'}}{r_{o'} - \rho_{o'}} = \frac{n'^2\,p}{e};$$

and, since, by (U)

$$r_o - \rho_o = r_{o'} - \rho_{o'}$$

it follows that

$$\frac{r_o\ \rho_o}{r_{o'}\ \rho_{o'}} = \left(\frac{n}{n'}\right)^2 \qquad (V)$$

From Equations (U) and (V) the following useful relations are obtained

$$r_{o'}\ \rho_{o'} = \left(\frac{n}{n'}\right)^2 r_o\ \rho_o$$

$$r_{o'} + \rho_{o'} = \pm\sqrt{(r_o - \rho_o)^2 + 4r_o\ \rho_o\ (n'/n^2)}\ ]$$

$$q = \tfrac{1}{2}\ [\,r_o + \rho_o \pm \sqrt{(r_o - \rho_o)^2 + 4r_o\rho_o\ (n'/n^2)}\ ]$$

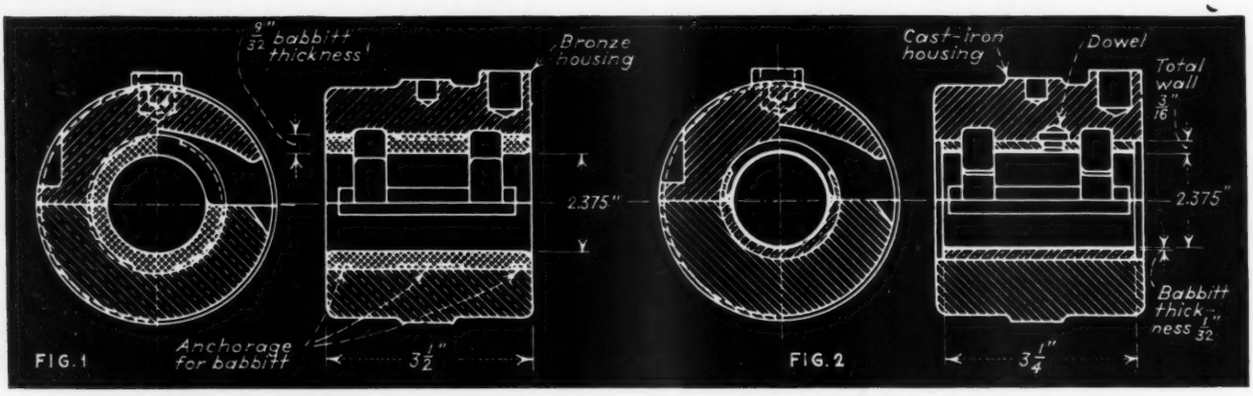
where the sign + or — is chosen according to whether the engagement of L and L' is external or internal.

Where L is the one-lobed curve, or ellipse with center of rotation at one focus,

$$r = \frac{(2r_o\ \rho_o)\ /\ (r_o + \rho_o)}{1 - (r_o - \rho_o)\ (\cos\theta)\ /\ (r_o + \rho_o)} = \frac{p}{1 - e\cos\theta}$$

Hence, by analogy, p_n and e_n, in Equation (S), may be termed the "parameter" and the "eccentricity" of the n-lobed elliptical curve. Geometrically, p_n is the radius vector in L bisecting the angle included by adjacent maximum and minimum radii vectors. The p_2 of a two-lobed curve, which is the ellipse with center of rotation at the center of figure, should not be confused with the parameter proper of that ellipse.

QUESTION AND COMMENT



Figs. 1 and 2—Babbitt-lined bronze housing in Fig. 1 requires more than 2 lb. of bearing material for replacement. Redesigned bearing in Fig. 2 has bronze-back babbitt bearing pressed into cast iron housing, the babbitt lining being only 1/32 in. thick

Replacement of Babbitt-Lined Bearings

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Chief Engineer, Federal-Mogul Corporation

● The so-called "home-made" babbitt-lined bearing is usually made by pouring the babbitt into a cast iron or bronze shell, as in shaft hangar bearings. Because babbitts do not bond well with cast iron, means such as grooves or dovetails are usually provided for anchoring the babbitt. Certain bronzes used for bearing housings particularly those containing a zinc content of 3 per cent or more, do not bond well with babbitt and also require some means for anchoring the babbitt. In such bearings the lining is approximately $\frac{1}{4}$ in. thick and when the metal required in the anchorages is added to this, the cost of material increases.

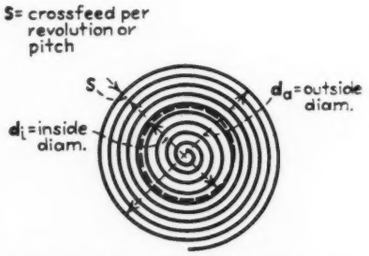
In Fig. 1 is shown the construction of a babbitt-lined bearing lubricated by a pair of oiling rings. The babbitt, which is 9/32 in. thick, is applied to a bronze housing and anchorages for retaining the babbitt are provided. Approximately two lb. of babbitt is used at an approximate cost of 97 cents per bearing. A bearing failure will cause loss of time for repair, as the bronze housing must be re-babbitted and re-bored.

A redesign is shown in Fig. 2 to permit the use of a precision type of bronze-back, babbitt-lined bearing. The babbitt is 1/32 in. thick. Because of the smaller amount used, the cost of the babbitt is reduced to approximately 35 cents. The bronze back is heavy enough to provide a proper stiffness factor and the bronze analysis is such that a good bond is obtained. Because of the uniform wall thickness, the babbitt structure can be controlled to obtain the maximum load-carrying ability. The housing is made of cast iron as compared to bronze formerly used thereby effecting a saving in material cost. Should such an installation require replacement, a new bearing can be installed without further machine work, scraping or fitting.

Facing With Hydraulic and Mechanical Drives

DR. ENG. M. KRONENBERG

● Supplementing the article "Comparative Test Results of Gear and Hydraulic Drives," which appeared on page 326 of the September 1936 issue of *Product Engineering*, the author derives a formula for determining the theoretical time of lathe facing operations when using a constant cutting speed. This constant cutting speed for facing operations, obtainable by the Lauf-Thoma hydraulic drive, is compared to a mechanical drive whereby a constant number of revolutions are required for facing a given area.



When facing in a lathe, cutting-edge of tool follows path of Archimedean spiral

The cutting-edge of the tool follows an Archimedean spiral, as shown in the illustration, when the transverse feed per rev. is constant. The length L of the Archimedean spiral is:

$$L = \pi S Z^2$$

When
S = feed per revolution
Z = number of coils or turns (1)

Furthermore if d_a is the largest diameter to be faced then the number of turns for facing from the center to the outer surface of diameter d_a will be:

$$Z = d_a/(2S) \quad (2)$$

Substituting Equation (2) in Equation (1):

$$L = \pi d_a^2/(4S) \quad (3)$$

Since the facing-time T_F in minutes, when the cutting speed v is constant, equals the length of the spiral divided by the cutting-speed, and with n_a r.p.m. at the beginning of the facing operation,

$$T_F = \frac{L}{v} = \frac{\pi d_a}{4 S d_a \pi n_a} = \frac{d_a}{4 S n_a} \quad (4)$$

In the case of mechanical drive

where T_m is the time in minutes, the formula for facing from the outside diameter to the center with the number of revolutions n_a constant is:

$$T_m = d_a/(2\,S\,n_a) \qquad (5)$$

A comparison of Equations (4) and (5) shows that facing with a contant cutting speed (obtainable by a hydraulic drive) requires only half of the time necessary to face with a constant number of revolutions (mechanical drive).

In practice it is impossible to obtain this reduction in time when facing to the center with constant speed, since the speed at the small diameters would have to be infinitely great, and the r.p.m. are necessarily limited. The cutting speed therefore decreases in the last part of the facing operation. When calculating the constant cutting speed to be used for facing a ring with outside diameter d_a and an inside diameter d_i, the equation is:

$$T_F = \frac{\pi\,(d_a^2 - d_i^2)}{4\,S\,v} = \frac{d_a^2 - d_i^2}{4\,S\,d_a\,n_a} \qquad (6)$$

The corresponding formula for facing the ring when the r.p.m. of face plate is constant becomes:

$$T_m = \frac{(d_a - d_i)}{2\,S\,n_a} \qquad (7)$$

Dividing Equation (6) by (7) shows the ratio of time required for facing a ring on a hydraulically driven lathe with a constant cutting speed and the time required for the same operation on a mechanically driven one with a constant number of revolutions, if the r.p.m. of face plate at the beginning of cut is the same in each instance. Thus:

$$\frac{T_F}{T_m} = \frac{d_a + d_i}{2\,d_a} \qquad (8)$$

From Equation (8) it can be deduced that when facing with a constant cutting speed on a hydraulic lathe, a saving of 25 per cent can be made if the smallest diameter d_i is one half of the largest diameter d_a; the saving is 37 per cent if d_i is one-quarter of d_a; and it is 45 per cent when d_i is one-tenth of d_a. These percentages will of course decrease if the speed is changed at different diameters by the workman on the mechanical machine.

What Type of Bearing Is Most Suitable?

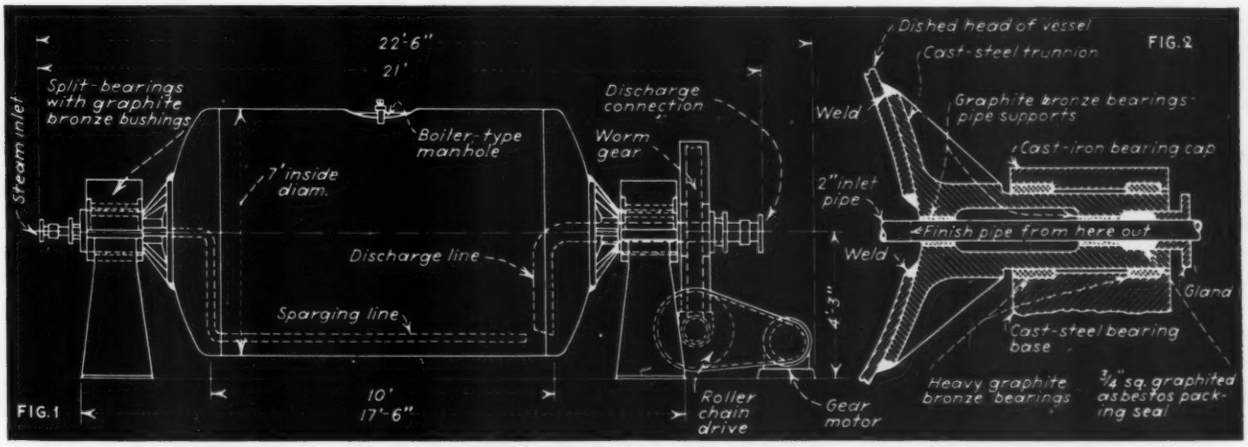
To the Editor:

● In the design of carbonator type digesters, as shown in Fig. 1, used in chemical plants, numerous problems arise for which there seems to be no general information available. This is especially true of allowable bearing pressures to use for conditions involving large diameters, high loads, temperatures up to 500 deg. F., and very low speeds.

As an example of such problems the cylinder shown in Fig. 1 is 7 ft. inside diameter by 10 ft. in length, exclusive of the dished head at each end, and revolves at one 1 r.p.m. in cast steel, hollow trunnions which are welded to the heads. These trunnion shafts are fitted with suitable seals and stuffing boxes for a steam sparger or sprinkling line and a steam discharge line as shown. The cylinder shell weighs 7,600 lb., the two standard dished heads weigh 6,025 lb., the two trunnions 3,960 lb., and the charge or contents of the cylinder weighs 25,000 lb., making a total of approximately 42,000 lb. that must be rotated.

During operation, in this particular installation, steam at 150 lb. gage (365 deg. F.) is sparged into the charge to bring the material up to temperature and to provide the necessary heat for reaction. The steam supply is automatically controlled by a thermostatically operated valve in the steam line. The material is thoroughly agitated by six flights or baffles 9 in. high, running lengthwise inside the cylinder. These flights are not bent nor curved but run parallel to the axis. Discharge is accomplished by merely opening the valve in the discharge line and allowing pressure inside the cylinder to force the contents through the discharge pipe.

In Fig. 2 is shown the design of the trunnion and bearing arrangement as used on either end of the cylinder. Although the illustration shows an inlet of 2-in. extra heavy pipe, the discharge end has a 6-in. extra heavy pipe, passing through the trunnion. These pipes are machined on the outside where they pass through the bearings to provide a smooth surface for the bearings and the stuffing boxes, the latter for sealing the pipe in the trunnion. As shown, the trunnion is welded both at the outside and inside of the dished head. Although radiation tends to dissipate heat from the bearings, some heat is generated in the bearing so that a temperature of 365 deg. F. may be considered constant. Although the question arises when to water-cool the bearings, it has not been necessary to dissipate heat by water-cooling when graphite-bronze bearings are used. Because of the large diameter of the trunnions, the load per sq.in. of projected area in this particular installation was only 150 lb. per sq.in., but it was found that the bearing was too short for



Drum of carbonator type digester having contents of 25,900 lb., or total revolving load of 42,600 lb. rotates 1 r.p.m., being driven by reducing gear motor, chain and worm wheel. Steam at 150 lb. gage is sparged into charge through bearing shown in Fig. 2 welded to each dished head. Temperature in bearing at times reaches 500 deg. F. Graphite bronze used

sufficient rigidity and therefore one bushing was placed at each end of the housing which is 18 in. long. I have also been unable to find what friction coefficient to use under these conditions and am therefore wondering if some of the readers of *Product Engineering* have any suggestions not only as to the design of the bearings but for the trunnions as well.

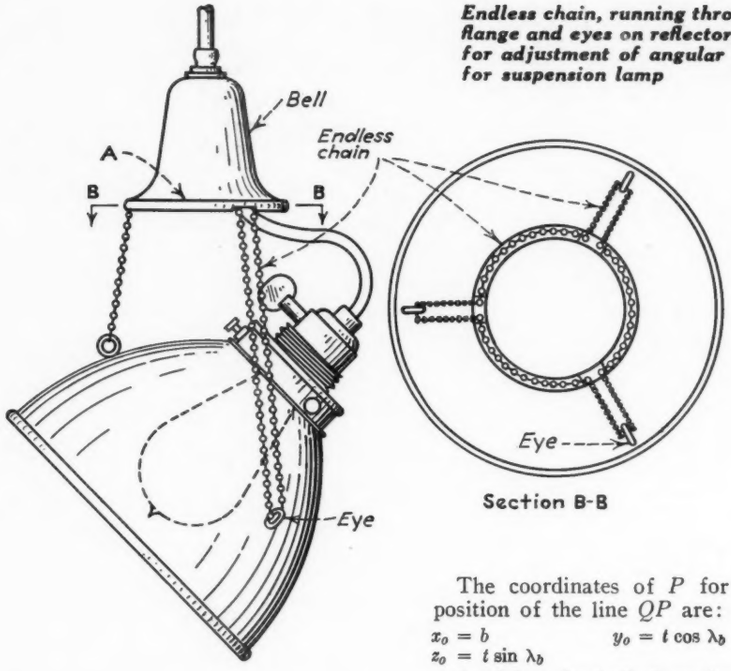
—HALE D. SEYMOUR
Mt. Vernon, Ill.

Lamp Suspension Gives Angular Adjustment

To the Editor:

● When the rays of a lamp are to be directed on a given object, it can be done readily if the lamp is built-in to form an integral part of the machine. But when the lamp fixture is suspended from the ceiling on an electric cord, it must be tied up in such a manner as to focus the light where it is required. In the accompanying illustration is shown a patented German method for tilting the lamp reflector to any desired angle.

The electric lamp is connected to the reflector in the conventional manner and is suspended from a bell, the lower end of which, at *A* is spun over to form a circular tube. An endless chain is guided in and out of apertures in the bell tube passing through three eyes fastened to the external surface of the reflector. The path of the chain is shown in the plan view.



Endless chain, running through bell flange and eyes on reflector, is used for adjustment of angular position for suspension lamp

Adjustment of the reflector can be made by using only one hand, the chain sliding through the eyes and tubular rim to accommodate any angular position. As the reflector is in balance at all times and because of the slight, but sufficient friction of the chain in the sliding members, the reflector can be quickly positioned by merely giving it a push.

—F. LOEWENBERG
New York, N. Y.

Axial Curvature of the Involute Helicoid

JOHN G. URBANIK
Rochester, N. Y.

● Although helicoidal surfaces of uniform lead have been used as machine elements for many years, the practical application of differential geometry for determining such surfaces has not been carried out very extensively. Aside from the generally used analysis of rectangular coordinates for surfaces, the development of vector analysis provides a most useful method in studying surfaces. Sometimes, however, it is possible to determine the curvature properties of surfaces by the use of the theorems of Euler, Meunier, Dupin and others if the principal curvatures can be obtained by kinematic or other methods.

Although the analysis given here for the axial section of the involute helicoid is in the conventional rectangular coordinate system, it has been patterned after the vector analysis solution. The usefulness of the vector method is that with slight modification, it can be used to find exactly the lines of contact between involute or screw helicoid worms and their mating worm gears.

As an example, it is required to find the curvature in an axial section of the involute helicoid, that is, the outline of a lathe tool, which if placed on center and given the proper lead will generate a mathematically correct involute helicoid. It is known that the involute helicoidal surface can be generated by a straight line rolling on a base cylinder. As shown in Fig. 1, let

b = radius of base cylinder
λ_b = base lead angle
$t = QP$, a parameter determining a point on the generating line
θ = parameter for the helicoidal motion

The coordinates of P for a fixed position of the line QP are:

$x_o = b \qquad y_o = t \cos \lambda_b$
$z_o = t \sin \lambda_b$

Applying the helical motion to this line as in Fig. 2, the lead of this motion is equal to $2\pi b \tan \lambda_b$, therefore

$$x = x_o \cos\theta - y_o \sin\theta = b\cos\theta - t\cos\lambda_b \sin\theta \quad (1)$$

$$y = x_o \sin\theta + y_o \cos\theta = b\sin\theta + t\cos\lambda_b \cos\theta \quad (2)$$

$$z = z_o + b\,\theta \tan\lambda_b = b\,\theta\tan\lambda_b + t\sin\lambda_b \quad (3)$$

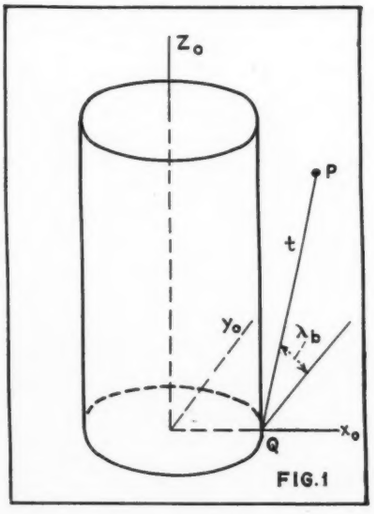


FIG. 1

As an equation of an axial section, let $y = 0$, all axial sections being identical except for position.

From Equation (2)

$$t = \frac{-\,b\tan\theta}{\cos\lambda_b}$$

Equations (1) and (3) then become:

$x = b/\cos\theta$
and
$z = b\tan\lambda_b(\theta - \tan\theta)$

These two equations are then the parametric equivalent of an axial section, but since $\tan\theta \gtrless \theta$, the sign of

the term z is arbitrarily changed.

$$x = b/\cos\theta \quad (5)$$

$$z = b\tan\lambda_b(\tan\theta - \theta) \quad (6)$$

By differentiating:

$$dx = (b\sin\theta/\cos^2\theta)d\theta \quad (7)$$

$$dz = b\tan\lambda_b\tan^2\theta\, d\theta \quad (8)$$

Therefore the inclination of the tangent line is:

$$dx/dz = 1/(\tan\lambda_b\sin\theta)$$

Let

$$dx/dz = \tan m$$

Then

$$\tan m = 1/(\tan\lambda_b\sin\theta) \quad (9)$$

To determine the radius of curvature,

Let

ρ = radius of curvature

Then by definition as shown in Fig. 3,

$$\rho = \frac{dz/\cos m}{dm} \text{ or } \rho = \frac{dz}{d\theta}\,\frac{d\theta}{dm}\,\frac{1}{\cos m} \quad (10)$$

From Equation (9):

$$\frac{d\theta}{dm} = -\frac{\tan\lambda_b\tan\theta\sin\theta}{\cos^2 m} \quad (11)$$

From Equation (8):

$$\frac{dz}{d\theta} = b\tan\lambda_b\tan^2\theta \quad (12)$$

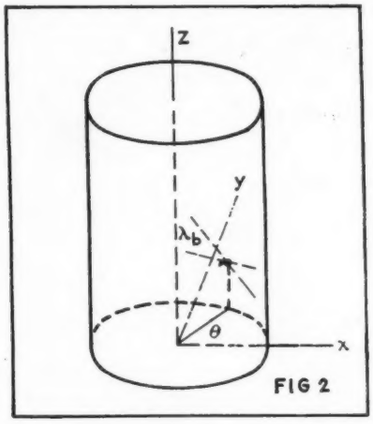


FIG 2

Therefore using Equation (10), after some simplification (and neglecting the sign of ρ):

$$\rho = \frac{b\tan\lambda_b}{\tan m\cos^3 m\,(\tan^2\lambda_b\tan^2 m - 1)^{3/2}}$$

In an axial section, we are interested in the curvature at the pitch point, therefore at a point the axial pressure angle ϕ_x of which is known.

Since

$$\phi_x = 90\text{ deg} - m,$$

$$\rho = \frac{b\tan\phi_x}{\tan^2\lambda b\cos^3\phi_x[1 - (\tan^2\phi_x/\tan^2\lambda_b)]^{3/2}} \quad (13)$$

The radius of curvature ρ in terms of the base cylinder radius b, the base lead angle λ_b, and the axial pressure angle ϕ_x, is therefore determined. For purposes of calculation, the following transformation of the formula is simpler to use:

R = pitch radius
ϕ_x = axial pressure angle
ϕ_t = transverse pressure angle

$$\rho = \frac{R}{\sin\phi_x\cos\phi_x\cos^2\phi_t}$$

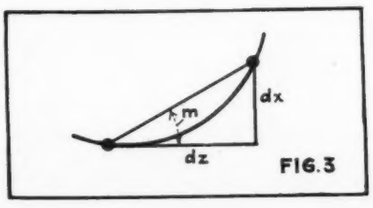


FIG. 3

Resiliently Mounted Motor Is Adjustable

To the Editor:

● Not only must the motor of the Speed Queen clothes washing machine made by the Barlow & Seelig Manufacturing Company be mounted resiliently to avoid vibration and to reduce noise, but motor adjustability must also be provided. This adjustability is necessary for maintaining proper belt tension as the motor is belted directly to the gear box.

To simplify design and for ease of assembly, the motor is hung vertically on the machine frame by four studs screwed into the motor flange, as shown in the illustration. Eight sandwich type rubber mountings are used, four on each side of the sheet metal motor plate. The machine frame (not shown) lies between this plate and the four upper sandwiches. The frame has four elongated holes through which the motor studs pass and on which loose-fitting nuts are used. The nuts are tightened sufficiently to provide the necessary resiliency of the motor yet with sufficient freedom so that the adjusting screw for belt take-up can pull the motor assembly along the frame. To provide the correct degree of compression in the sandwiches, serrated thread locks which grip at the roots of the threads, lock the hex nuts after they have been tightened.

—F. A. WESTBROOK
Center Conway, N. H.

Resilient mounting in which motor is adjustable

Can You Work This One?

H. E. SMITH
Newark, N. J.

This Month's Problem—

Rope, Pulley and Weight

A rope, the weight of which may be neglected, hangs over a smooth pulley in which friction may be neglected. One end is fastened to a weight of 150 lb. and the other end to a sailor weighing 150 lb., the sailor and the weight hanging in the air at the same elevation. The sailor begins steadily to climb the rope. Will the weight move at all, and if so will it rise or fall. Whichever it does, if it does either, how long and how far will the process continue?

Solution to Last Month's Problem—

How Much Money Had Each

Let x be the number of pennies each had at first.

No. (3) received x, took out $(2 + 4)$, and put in $x/2$; so that the sack then contained $[x\ 3/2 - 6]$.

Substitute (a) for 3/2

No. (5) received $(x\ a - 6)$, took out $(4 + 1)$, and put in enough to multiply by a, the contents of the sack when he received it. The sack then contained $(x\ a^3 - 6\ a - 5)$.

No. (2) took out $(1 + 3)$, and handed on $(x\ a^3 - 6\ a^2 - 5\ a - 4)$.

No. (4) took out $(3 + 5)$, and handed on $(x\ a^4 - 6\ a^3 - 5\ a^2 - 4a - 8)$.

No. (1) put in 2. The sack then contained 5 x.

Hence:

$$x\ a^4 - 6\ a^3 - 5\ a^2 - 4a - 6 = 5\ x$$

$$x = \frac{6\ a^3 + 5\ a^2 + 4\ a + 6}{a^4 - 5}$$

$$x = \frac{2[(6\times3^3) + (5\times3^2\times2) + (4\times3\times2^2) + (2^3\times6)]}{3^4 - 5\times2^4}$$

$$x = \frac{2(162 + 90 + 48 + 48)}{81 - 80}$$

$x = 696$ pennies each had at first.

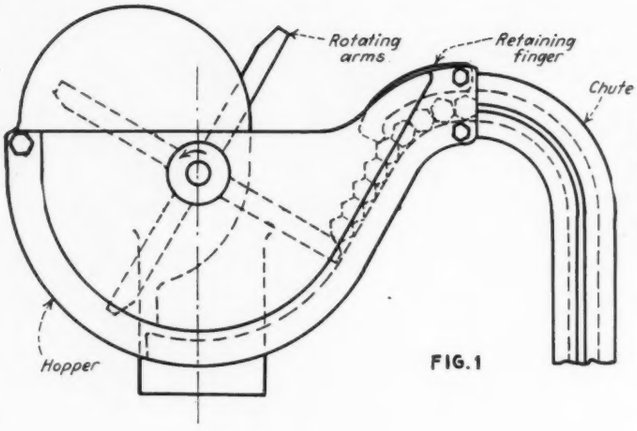


FIG. 1

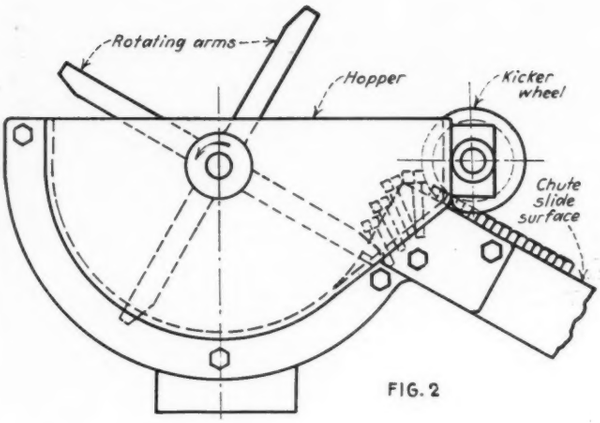


FIG. 2

Hoppers for Feeding

W. E. SUMMERHILL

Designs of stationary hoppers with rotating arms and disks, reciprocating and oscillating blades, and those wherein the hopper rotates angularly or is automatically tilted

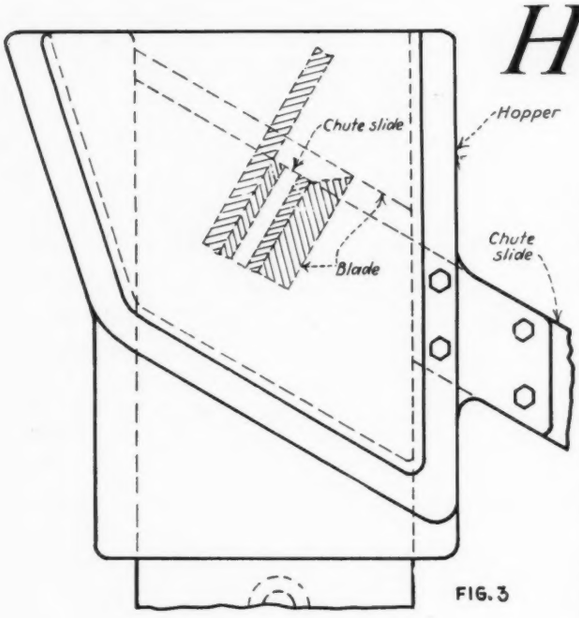


FIG. 3

Fig. 1—Rotating arms of nut hopper push nut blanks up incline into chute. Retaining finger holds several nuts and prevents them from sliding back into the hopper.

Fig. 2—Same type hopper and rotating arms as in Fig. 1, but different chute, designed to feed bolts. Kicker-wheel at mouth of chute kicks back into hopper the bolts that do not enter the chute properly.

Fig. 3—Reciprocating blade, the upper edges of which are on the same angle as chute slide, moves vertically from bottom of hopper to top of chute. Blade is bevelled 45 deg. at top edge to help tumble bolts into slot.

Fig. 4—Hopper used for feeding shell-like pieces into tube conveyor. A reciprocating plunger picks up work at lower end of stroke and deposits it in snap-fingers at end of conveyor tube.

Fig. 5—Hopper is adjustable for feeding various lengths and diameters of plain round stock, the pieces falling into chute by

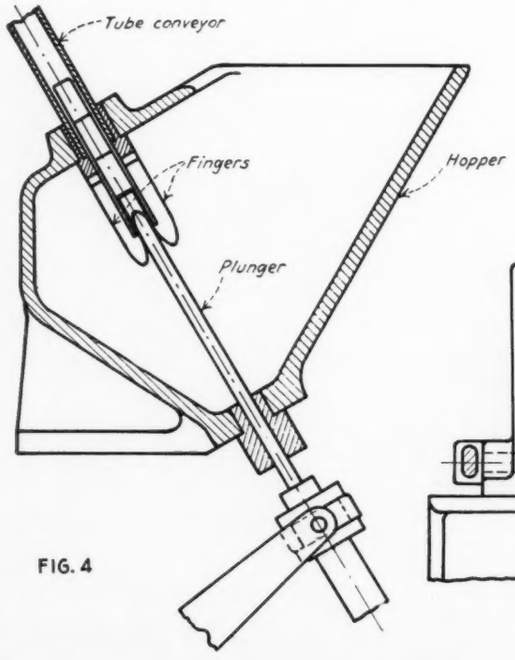
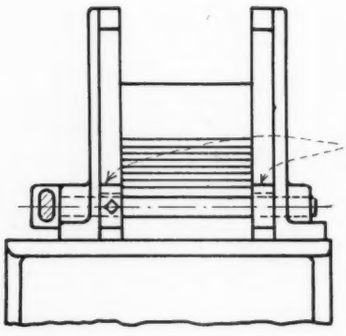


FIG. 4



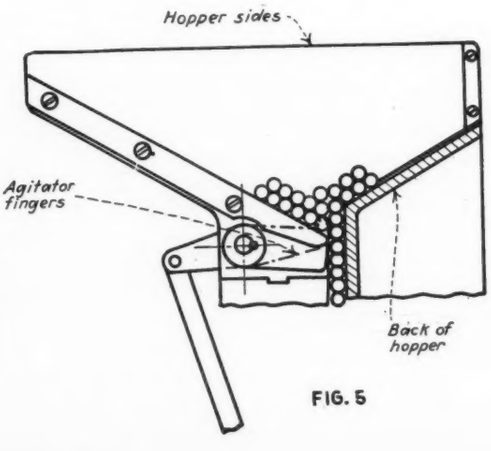


FIG. 5

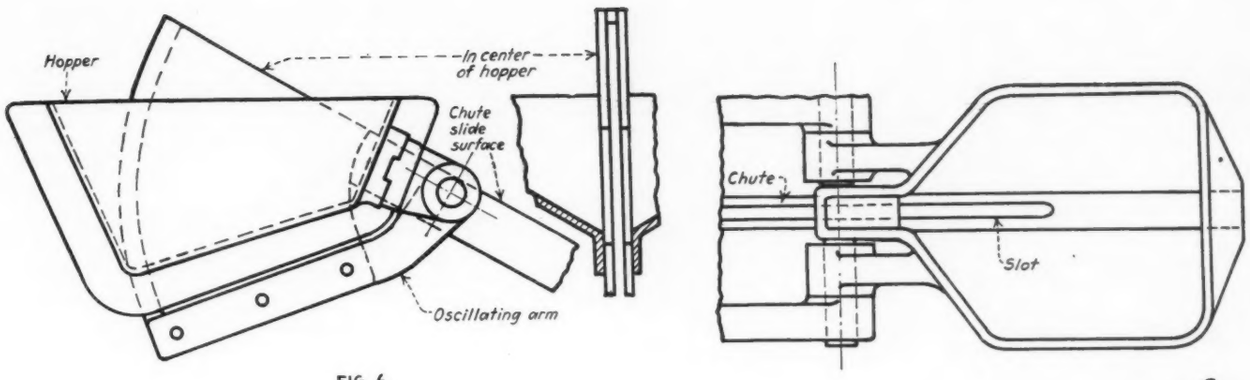


FIG. 6

Work Automatically

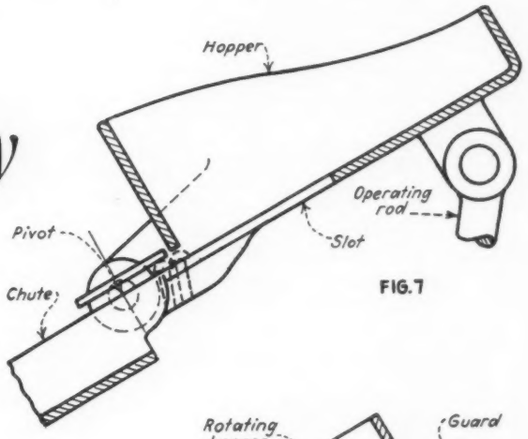


FIG. 7

gravity. Agitator finger at either end of work prevents bridging or wedging of blanks over chute opening.

Fig. 6—An oscillating arm carries the blade through the center of the bolt hopper and at the top of its stroke forms a continuation of the bolt chute. Sides of hopper are inclined toward center to feed bolts into blade at low position in hopper. One blade is used for each diameter of stock handled, tapered spacers in hopper being adjustable to accommodate varying widths of blade.

Fig. 7—Tilting hopper for small rivets and screws in which the work falls into slot at bottom center of the hopper which is tilted to same angle as chute.

Fig. 8—Rotating hopper set at angle is slotted at lower face to feed into the chute small cup shaped objects, as shown at *A*, positioning them with their open end up. Should cups enter chute open end down they will drop through selector slot in chute thereby allowing only those correctly positioned to proceed to assembly point.

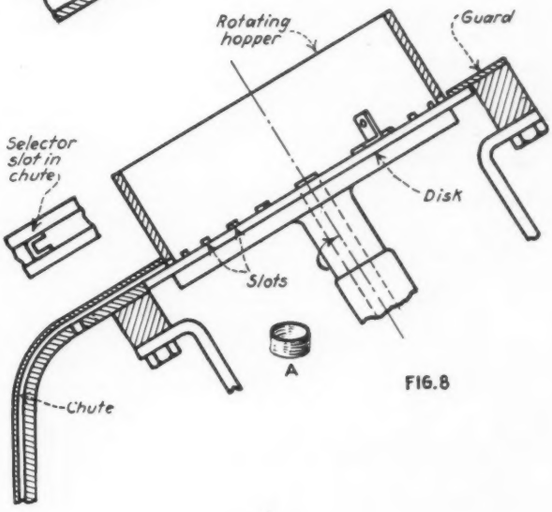


FIG. 8

Fig. 9—Vertical rotating disk hopper for feeding shouldered pieces to chute. By adjusting hardened dog-point screws it is possible to feed pieces with difference of only 0.010 in. on the diameter.

Fig. 10—Another type of vertical rotating disk hopper for feeding hollow cylindrical pieces having blind hole. Prongs are milled on periphery of disk which prevent work being fed open end up into chute.

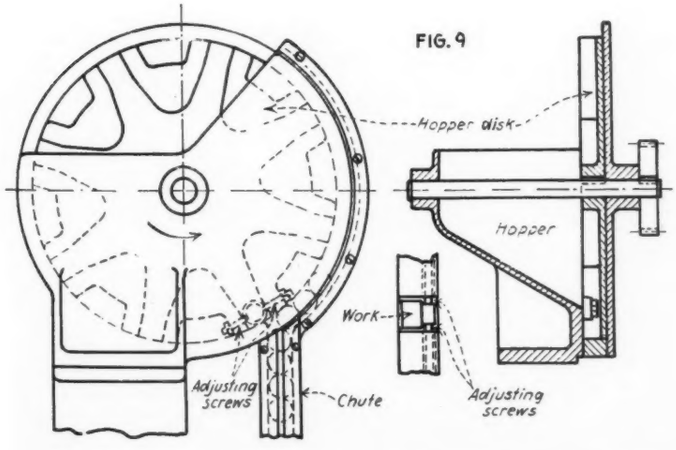


FIG. 9

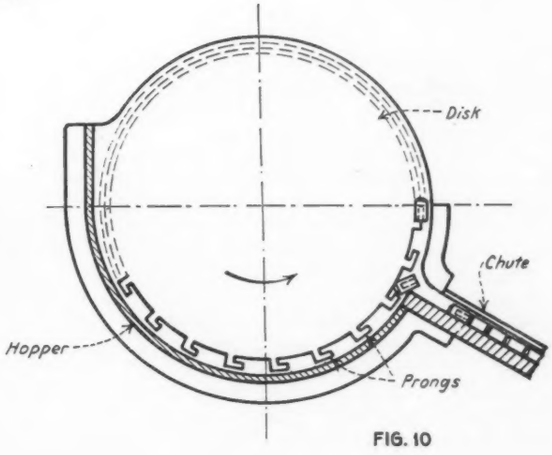


FIG. 10

NEWS

Porcelain Enamelers Announce Conference

PRELIMINARY plans have been announced for holding an "Enamelers Round Table Conference," sponsored by the Porcelain Enamel Institute, to be held next May at the University of Illinois. F. E. Hodek, Jr., of the General Porcelain Enameling & Manufacturing Company, who is vice-president of the institute, is in charge of this new project which is planned to become an annual affair.

According to the initial announcement the conference will be a three-day affair and will include discussions of enameling problems of interest to the entire industry. In addition to the general sessions, which will be all-industry round table discussions, the conference will be divided into sectional groups which will discuss problems pertaining to the enameling of cast iron, sheet metal and hollowware. Prof. R. M. King of Ohio State University and Dr. A. I. Andrews of the University of Illinois will take prominent parts at the meetings.

Enamelers forums and short courses formerly conducted by Ohio State University, the University of Illinois and the Ferro Enamel Corporation are to be discontinued in favor of this Institute conference.

Oil Burner Show Extends Its Scope

BECAUSE of the close relationship between oil burners and air conditioning, the scope of the 1937 Show or Exhibit sponsored by the American Oil Burner Association will include not only oil burning equipment and accessories, but also air conditioning equipment. This National Oil Burner and Air Conditioning Exposition and Convention will be held March 15 to 19 at Convention Hall of the Commercial Museum in Philadelphia. Among the exhibitors expected to participate are manufacturers of domestic, commercial and industrial oil burners, distillate burners and air-conditioning equipment for heating, cooking and power purposes. Numerous operating exhibits will be included and some of the leading oil companies will show several kinds of fuel oil which they furnish for particular domestic and industrial uses.

Among the accessory group will be included manufacturers of automatic controls, boilers, pumps, tanks, gages and other essentials.

Convention sessions will be held at Convention Hall, adjacent to the exhibit. The program, still in tentative form, will include conferences open to all burner manufacturers, technical sessions covering detail pnases of oil burner and air conditioning industries, group meetings of accessory manufacturers, general marketing sessions and dealers conferences. In addition, there will be numerous meetings to which the public will be invited, these meetings covering in detail the ramified parts played by oil as a fuel in domestic heating, cooking and air-conditioning.

Acoustics Standards Are Now Available

THE American Standards Association, 29 West 39th Street, New York City, has recently announced a schedule of quantity prices for copies of the American Tentative Standards for Acoustical Terminology (Z24.1-1936), Noise Measurement (Z24.2-1936) and Sound Level Meters (Z24.3-1936). The price of one copy of any one of these standards is 25 cents, the remittance to be enclosed with the order. For quantity orders of any one of the acoustical standards or for any combination of the three, 10 to 24 copies can be obtained at 20 cents per copy, 25 to 99 copies at 15 cents per copy, and 100 to 249 copies at 10 cents per copy. For 250 copies or more, the price is 8 cents per copy.

New Name for American Chain

AMERICAN Chain & Cable Company, Inc. is the new name of the company previously known as American Chain Company, Inc. This company has long been recognized as the world's largest manufacturer of chains for all purposes and since 1924 has become a leader also in the production of wire rope or cable. It was this fact which caused the directors of the corporation to enlarge the name to American Chain & Cable Company, Inc. In addition to chain, wire rope and cable, this company also manufactures malleable castings, railroad specialties, abrasive cutting machines, nibbling machines, automotive service station equipment, Page fence, wire and rod products, traffic tape, welding wire, preformed and non-preformed wire rope, bronze, iron and steel valves, electric steel castings, and numerous other products.

New Welding Foundation

DEDICATED to the encouragement of study and research for the benefit of the arc welding industry, a significant new Foundation has just been established by a vote of The Lincoln Electric Company's directors. This new fund has been named "The James F. Lincoln Arc Welding Foundation," in honor of the pioneer work of the company's president in promoting arc welding and in perfecting and developing arc welding equipment and electrodes.

One of the primary functions of this new foundation will be the stimulation of original design, in order that arc welding processes may be more widely utilized in modern fabrication and engineering design. Direction of the Foundation's work will be under Dr. E. E. Dreese, head of the Department of Electrical Engineering at The Ohio State University. Other trustees who will serve with Dr. Dreese on the Foundation are W. B. Stewart, distinguished member of the Cleveland Bar, and H. R. Harris, vice president of the Central National Bank of Cleveland.

Dr. Dreese is an outstanding member of the American Institute of Electrical Engineers, and has wide connections in the field of general science. His important position with the Foundation places its program on the highest plane and assures the carrying out of studies which will view arc welding with full regard to both its economic and social significance.

In commenting on the establishment of this Foundation, president James F. Lincoln stated:

"The sole purpose of our new Foundation is to stimulate the ingenuity of scientists, engineers and skilled workers interested in extending the frontiers of knowledge and achievement. This arousing of interest will result, I believe in lower production costs for thousands of devices and commodities now employed in manufacture or moving in commerce; and

I feel signally honored in having my name attached to a movement which seeks to give constantly increasing markets the benefit of better and more serviceable goods."

It is expected that the directors of the Foundation will soon formulate a comprehensive program, and will announce the first of their activities at an early date.

Industrial Color Service by Sherwin-Williams

FORMATION of an Industrial and Transportation Color Service, under the direction of their Mr. R. H. Hookway, has just been announced by the Sherwin-Williams Company. This new department combines its efforts with those of the Sherwin-Williams Paint Engineering Service to offer designers, manufacturers and industrial consultants assistance not only in developing desired colors and color combinations for "dressing up" new or old products, but also gives information on types of finishes and their suitability for specific application requirements. The service applies to all types of industry and transportation to aid industrial designers and engineers in securing effective color treatments and suitable types of finishing materials. After the color scheme has been worked out, panels and paint-out tests are made on the material to be finished, to assure the desired qualities, namely, durability, degree of gloss and color retention.

Colorist for the Sherwin-Williams Company, Mr. Hookway and his department have cooperated with industrial designers and manufacturers on numerous special problems of color and finishes on such well-known jobs as The New Haven Streamlined Comet, The New York Central Mercury, other railroad trains, ferry boats, automobiles, tractors and household equipment.

Consider Standard Welding Fittings

ACCORDING to the Prime Movers Committee of the Edison Electric Institute, the lack of standardization in the dimensions of welding fittings of various makes has caused many difficulties, and that committee has asked the American Standards Association to consider whether the standardization of these fittings could be undertaken under ASA procedure. In making their request, the committee reported:

"A typical case of lack of uniformity in dimensions is illustrated by the 4 x 3 in. concentric welding reducer which is offered in four different lengths, depending on which of five manufacturers made the product. Other welding fittings involve similar wide differences.

"Where it is desired to use welding fittings, these conditions require that the vendor of the fittings be selected prior to detailing the pipe lines on the drafting board, otherwise the material received may not coincide with design details. . . .

"The project is particularly urgent in view of the increased extent to which welded construction is being used in power-plant piping. It is urged, therefore, that the dimensions of the more commonly used welding fittings be standardized to facilitate their use and enable their purchase on a competitive basis from several sources. In such a standardization program, other items should be considered in addition to dimensions, as: (1) details of weld chamfer; (2) materials; (3) tolerances."

This request has been brought to the attention of the American Standards Council, as reported in the December number of *Industrial Standardization*, and is now being given due consideration as to how it should be dealt with under ASA procedure.

Generic Term for Synthetic Rubber

IN ORDER to provide manufacturers with a name which can be applied both to their unvulcanized synthetic rubber and the products made from it, corresponding to the accepted usage of the term rubber, the E. I. du Pont de Nemours & Company has announced that the word neoprene has been adopted as a generic term for the chloroprene polymers formerly sold by that company under the trade mark DuPrene. The new name "neoprene" will not be trade marked and may be used freely by rubber manufacturers in describing products which display the distinctive characteristics of this unique material. The company also states that no change in the product has been made and that adequate steps will be taken to popularize this new name with the consumer public.

This synthetic rubber formerly marketed under the name DuPrene and now called neoprene is used in the manufacture of oil and gasoline hose, printing rollers, belting, packing, molded parts for the automotive and refrigeration industries and many other products where rubber-like properties combined with an exceptional stability against deteriorating effect of oils, gasoline, solvents and refrigerants is required. Neoprene is also of special value where heat, sunlight and ozone is encountered. The rapidly widening market for neoprene is indicated by the fact that since the completion of the second manufacturing unit in August, 1935, the sale of neoprene has increased six-fold.

Do You Know That—

- Carbon steels that have been tempered at low temperatures possess a comparatively low wear rate and the surfaces wear smooth. But when these steels are tempered above certain temperatures the rates of wear become extremely high and the worn surfaces are rough. (38)

- The two best color systems for signal lights for a six-color system are: (1) red, orange-yellow, white, green, blue, and purple; (2) red, orange, yellow, white, green, and blue, as determined by an investigation conducted by the Bureau of Standards. (39)

- A recently perfected process gives corrosion protection and paint adherence to magnesium alloys, equivalent to that obtained by the chrome pickle Because it does not reduce the cross-section of the material, the process can be used on machine parts. (40)

- Pipe made from a recently developed quality of open-hearth steel is now being made to meet the demands of the refrigerating industry where pipe is subjected to severe cold working. (41)

- Steel plates, 1 in. thick, can now be spot-welded together. The secret lies in the method of welder control. Similarly, it is now possible to join aluminum and copper rods by resistance welding. (42)

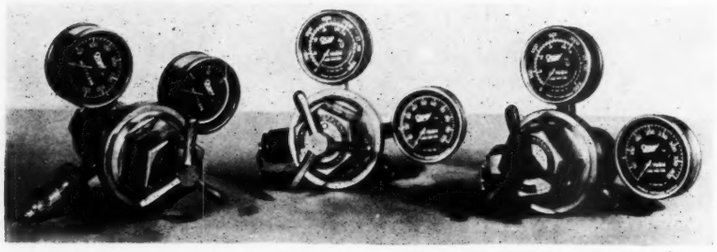
MEETINGS

National Oil Burner and Air-Conditioning Exposition and Convention—Commercial Museum, Philadelphia, Pa. March 15–19. Oil Burner Institute, G. Harvey Porter, managing director, 30 Rockefeller Plaza, New York, N. Y.

American Institute of Mining & Metallurgical Engineering—147th annual meeting. Feb. 15–18, Engineering Societies Building, 29 West 39th St., New York, N. Y.

Packaging Exposition—Pennsylvania Hotel, New York, N. Y. March 23–26. Roberts Everett Associates, show managers, 232 Madison Ave., New York, N. Y.

New Materials and Parts

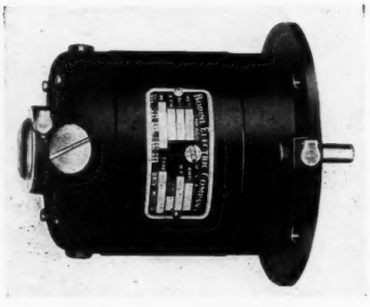


New Regulators

Designed to provide oxygen and acetylene regulation through the range of welding and cuting operations. They are of two-stage construction, valves are stem-operated and close with the incoming pressure. Bodies and caps are pressure forged. Point of the pressure-adjusting screw is held in alignment by ball bearings. Dials are colored with calibrations silvered. Linde Air Products Co., 205 East 42d St., New York, N. Y.

Flange Mounted Motor

Makes it possible to mount either horizontally or vertically with shaft perpendicular to the mounting surface. A groove is provided on the face of the flange to insure accurate location and a tight fit. Adaptable to either ball or sleeve bearing motors. Normally rated at 1/50 hp. but lower ratings can also be furnished in practically all types of windings regularly supplied on fractional hp. motors. Bodine Electric Co., 2254 W. Ohio St., Chicago, Ill.

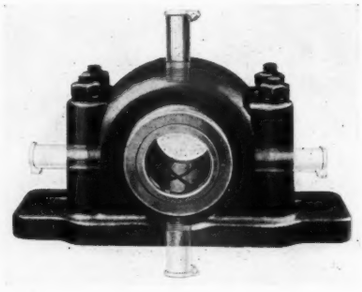


New Enamel

For obtaining chemical metal coloring effects without the use of chemical processes. It is semi-transparent when mixed with clear lacquers and is said to produce a transparent colored effect that is fast to light and durable. Suitable for compacts, lighting fixtures and novelties where a colored metallic finish is desired. Known as "Plate-lustre" and is supplied in an assortment of colors. Maas and Waldstein Co., Graybar Bldg., New York, N. Y.

Universal Pillow Block

After mounting the block in the position needed to suit the installation, the self-aligning bearing ball unit is turned so that the oil cup is in an upper quadrant. The oil cup is then screwed in place. A supply of fresh oil is contained in an upper reservoir which feeds the bearing as needed through plugs and graphite pressure-packed channels. Recovery grooves divert used oil into a lower wool-packed reservoir from which oil is again fed into the bearing. Randall Graphite Products Corp., 609-613 W. Lake St., Chicago, Ill.

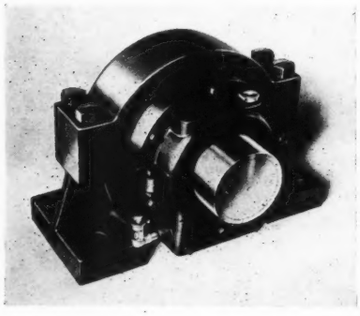


New Electrode

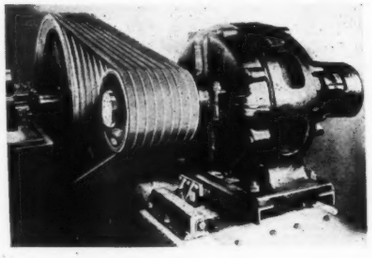
Designed for welding high strength steels. It is heavily coated and of the shielded arc type. Deposited in mild steel plate the weld metal is said to have a tensile strength of 100,000 lb. per sq.in., a yield point of 85,000 to 95,000 lb. per sq.in., and ductility of 12 to 18 per cent, in the as-welded condition. When stress relieved these values are greatly increased. "Shield-Arc 100" electrode is suitable for flat, vertical and overhead welding and is available in $\frac{1}{8}$, $\frac{5}{32}$ and $\frac{3}{16}$ in. sizes. The Lincoln Electric Co., Cleveland, Ohio.

Thrust Pillow Block

Has a separate heavy-duty ball bearing as an integral part of the assembly to take the thrust loads in addition to a roller-bearing for radial



loads. A two-part spherical housing provides self-alignment. It is oil-tight and equipped with a levelling device for oil lubrication. Available for shaft sizes from $3\frac{5}{16}$ to 10 in. and up to 250 tons radial load capacity. Fafnir Bearing Co., New Britain, Conn.

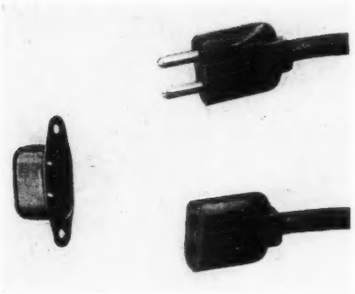


Pivoted Motor Base

Improves V-belt performance and maintains constant tension by automatically taking up belt stretch. Pivoting unit is mounted in adjustable side boxes with screw take-up. Further adjustment can be made by changing position of angle pieces in reference to rocker bearing casting. Unit is made in two other styles, one that is used for ceiling mounting and another having a vertical base. Rockwood Mfg. Co., 1801-2001 English Ave., Indianapolis, Ind.

Appliance Connector

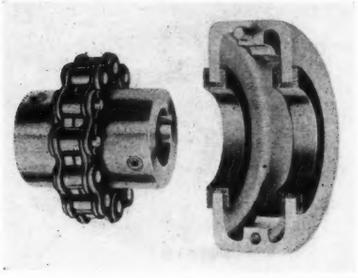
A metal receptacle designed for building into an appliance or light service machine for power connection. It is made to set in flush. An all-rubber connector molded into the



cord completes the assembly. Another combination is available in an all-rubber connector and plug. General Electric Co., Bridgeport, Conn.

Flexible Coupling

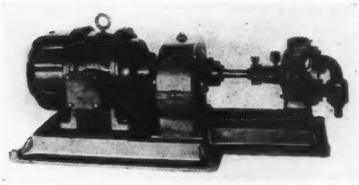
Has improved cover of balanced construction which grips the chain roll. The coupling is made with wide chain to obtain maximum bearing area. The shearing stresses are taken by combined cross section of the chain pin, bushing and roll. Lubricant is retained in cover by oil tight gaskets cemented to cover halves. Available in sizes from ½ to 8¼ in. diam. bore. The Whitney Chain & Mfg. Co., Hartford, Conn.



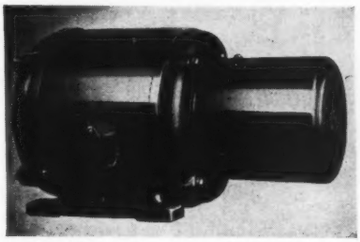
Flexible Shaft Handpiece

End of spindle is formed to receive flexible cable, and cap is threaded for flexible casing end fitting. The hardened steel spindle and ball bearing assemblies are protected from grit and chips by felt washers. Rounded nose piece is case hardened steel. Two interchangeable chucks are supplied, one for tool shanks up to 5/32 in. diam. and one up to ¼ in. diam. Speed 8,000 r.p.m. maximum. Overall length 5⅝ in., outside diam. 1¼ in. approximate. The S. S. White Dental Mfg. Co., 10 East 40th St., New York, N. Y.

Back-geared Pump Drive



Gear box is totally inclosed with the gears operating in a bath of oil. Reduction gears are made of steel for 2 hp. and greater. Standard equipment includes an end-thrust bearing for the rotor shaft, the bearing is of the flange mounting type and bolts directly on to the gear box. Pump speeds can be reduced to 100 r.p.m. from a 1,750 r.p.m. motor. Available from the 20 g.p.m. pump at 450 r.p.m. to the 90 g.p.m. pump at 390 r.p.m. Viking Pump Co., Cedar Falls, Iowa.

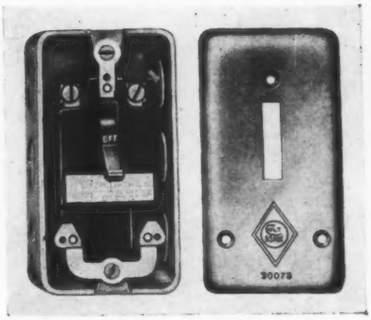


Motor Mounted Brake

Totally inclosed automatic solenoid brake can be supplied mounted by the motor manufacturer. Suitable for laundry, textile and woodworking machinery, hoists and machine tools. The rotating member is a heavy molded disk protected from dirt and injury by a case. Available for a.c. or d.c. service, from fractional hp. to 10 hp. at 3600 r.p.m., with dust-tight, weatherproof and water-tight inclosures. Cutler-Hammer, Inc., 12th & St. Paul Ave., Milwaukee, Wis.

Auxiliary Circuit Breaker

Designed to be mounted in standard wall boxes or in special cabinets for surface mounting. Two types are available, one a fixed capacity device, made in 10 differently rated sizes, and the other with interchangeable heater elements covering a range of 20 different ratings. Known as Yoke



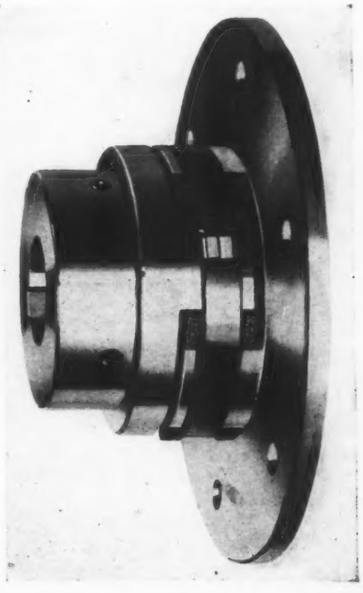
Mounted Protectit, both types have a 1-hp. and 250 volt a.c. or d.c. maximum ratings. Colt's Patent Fire Arms Mfg. Co., Hartford, Conn.

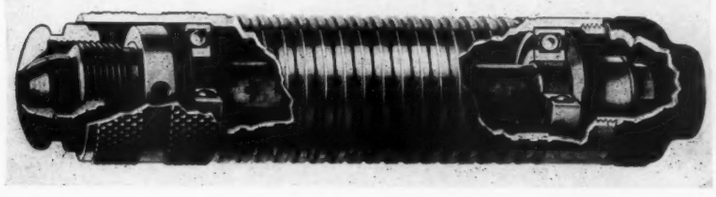
New Electrode

For use with small transformer type a.c. arc welders. Striking and maintaining an arc without difficulty, it is easy to handle and works with a.c. or d.c. machines. Slag interference is reduced to a minimum with removal simplified. New electrode is known as the Wilson No. 520. Wilson Welder & Metals Co., 60 East 42d St., New York, N. Y.

Compact Flexible Coupling

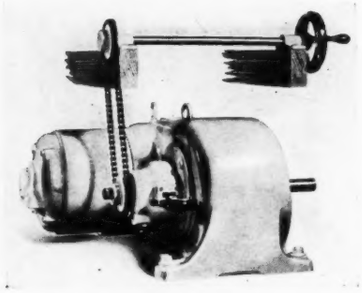
Flanged half bolts on to flywheel, clutch or brake drum. Overall length is reduced about a third allowing more space for other parts or for shortening shaft. Free floating load-cushions are always in sight and are held in place by a removable outside retaining-spring band. Types of cushioning material available are: a brake lining composition for heavy shock loads, leather for sustained loads with





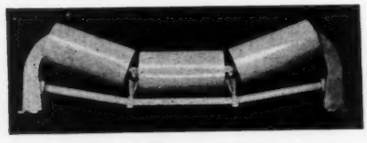
misalignment, and a rubber duck fabric for fluctuating loads where resilience is required. Made in sizes from 3 to 14 in. bores and capacities up to 2,500 hp. at 100 r.p.m. Lovejoy Tool Works, 5019 W. Lake St., Chicago, Ill.

Variable-Speed Control



Standardized parts for obtaining remote mechanical control for use with Varidrive motor. Parts consist of handwheels, bearings, shafting, sprockets and chains so that user can mount into an assembly to suit requirements. U. S. Electrical Motors, Inc., Los Angeles, Calif.

Truss Frame Carrier



Roller assembly is tilted to center the belt, outer rollers are inclined to give the belt a 20 deg. trough. Load is carried by a tension rod instead of a heavy cross member. All parts are made of steel and malleable iron. Rollers are built of 5 in. diam. steel tubing pressed over end plates. Renewable cartridge-type ball-bearings packed with grease and sealed at factory are used throughout. Built in sizes for 18 to 48 in. wide belts. Stephens-Adamson Mfg. Co., Aurora, Ill.

Explosion Tested D. C. Motors

Built and tested to withstand an internal explosion of methane or coal dust without injury to the equipment and without emitting flames or sparks which would ignite surrounding gas. A fan on the commutator end forces cooling air through passages. The patented double walled front bracket is equipped with four screw type hand hole covers for inspection and adjustment of brushes. Each cover has a hole in the web for insertion of either

a wire meter seal or chain and lock. Available in sizes from 5 to 75 hp. at 115, 230 and 550 volts. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Heat Indicating Paint

Changes color when exposed to heat. Used to detect serious deviations in temperature of processing equipment and machinery. Some colors turn back to original color when cooled, others will remain permanently changed. A series of five permanent changes and seven retroactive color paints are available; the former change at temperatures ranging from 300 to 734 deg. F., the latter from 104 to 464 deg. F., with a safety margin of 25 deg. These paints are prepared in types for exterior and for interior uses. The Efkalin Co., 804 East 141st St., New York, N. Y.

Temperature Control Unit

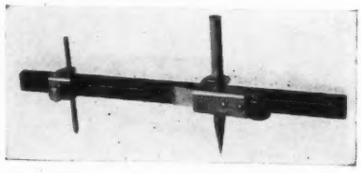
Uses a bimetallic element to actuate a mercury switch. The case of the mercury switch is made of metal and is sheathed in a plastic molded material. The housing is of one-piece construction finished in crackled art lacquer. The control units are designed for industrial and domestic applications, where accuracy is required on close operating differentials. Jefferson Electric Co., Bellwood, Ill.

Shock-resistant Plastic

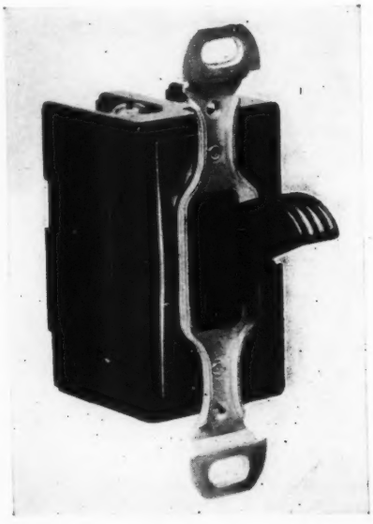
New plastic material suitable for heavy-duty parts and applications subjected to shock, is said to have three times the impact strength of standard compositions. Can be preformed with hand equipment. The molding can be machined and sanded subsequent to buffing without exposing filler spots or impairing the qualities of the material. Known as 1547 Black and weighs 22.4 grams per cu. in. General Plastics, Inc., 811 Walck Road, North Tonawanda, N. Y.

Beam Compass

Uses any standard pencil or ruling pen in the holder. After tightening the center pin holder to beam with finger screw, the pencil or pen holder is adjusted by propelling the roller along



the beam to suit radius desired. The metal parts are nickel silver. Beams 12, 24, and 36 in. long are available. A. P. Bartusch, 775 Walnut St., Lockport, N. Y.



Slow Break Switch

For industrial machines and appliances using a.c. fractional hp. motors, and special lighting circuits. Has heavy silver contacts to assure positive contact and long life. Toggle handle and housing are made of plastic molded material. Fits standard boxes. Can be furnished with boxes for exterior mounting. 10 amp. 250 v. rating. Hart Mfg. Co., Hartford, Conn.

MANUFACTURERS' PUBLICATIONS

Automotive Uses of Nickel Alloy Steels—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Bulletin U-1, 8 pages, 8½x11 in. Describes alloy constructional steels as associated with the development of motor vehicles and a number of specific applications.

Bakelite Molded—Bakelite Corp., 247 Park Ave., New York, N. Y. Booklet, seventh edition, 48 pages, 6x9 in. Contains a detailed description of Bakelite molded, its characteristics, tables of properties of various types of molding materials, and industrial applications. Also describing molding equipment and mold designs.

Bi-rotary Heater Switches—Hart Mfg. Co., Hartford, Conn. Folder, 8½x11 in. Describes range switch, construction of hydraulic oven thermostat, and two-heat slow-break heater switch.

Continuously Smelted Frits—The Porcelain Enamel & Mfg. Co., Baltimore, Md. Folder, 8½x11 in. Announcing a new method of smelting frits, the history of its development, and advantages derived from its use.

Craftsmanship in Metals—American Emblem Co., Inc., Utica, N. Y. 52 pages, 6x9 in. Describes the products of the company with illustrations of advertising novelties, gifts, premiums, medals, buttons, badges, name plates, escutcheons and stampings.

Data on Sicromo Steels—The Timken Steel & Tube Co., Canton, Ohio. Circular, 8 pages, 8½x11 in. Presenting additional information on Sicromo 1, 2, 3, and 5 steels, analyses are given, also discusses the effect of silicon and chromium on the oxidation resistance of steels.

Dilophane — Continental - Diamond Fibre Co., Newark, Del. Folder, 4 x 9 in. Describes and illustrates in color a translucent laminated synthetic resinous material, its physical and chemical properties, with some suggested uses.

Enduro Stainless and Heat-Resisting Steels—Republic Steel Corp., Republic Bldg., Cleveland, Ohio. Series of five booklets, 8½x11 in. The first is of general nature and illustrates many applications of Enduro; it contains a chart listing the analyses and properties of 13 types and contrasts them with carbon steel S.A.E. 1020. Another table sets forth the degree of corrosion-resistance of four leading types of Enduro in the presence of foods, chemical products and reagents. The other booklets feature data covering 18-8; straight-chromium types AA, S, S-1, and FC; heat-resisting types HCN, HC and NC-3; and 4-6 chromium steels respectively.

Felt Oil and Grease Seals—Gits Bros. Mfg. Co., 1859 S. Kilbourn Ave., Chicago, Ill. Catalog No. O.S. 36, 16 pages, 8½x11 in. Illustrates and describes various types, installations and mountings of oil and grease seals, contains tables of dimensions and recommended speeds.

Free Wheeling Clutch—The Morse Chain Co., Division Borg-Warner Corporation, Detroit, Mich. Folder, 6 pages 8½x11 in. Explains with text and illustrations construction and applications of free wheeling clutch, contains data tables of sizes, types and capacities.

Industrial Controls — Automatic Temperature Control Co., 34 E. Logan St., Philadelphia, Pa. Bulletin G-9, 8 pages 8½ x 11 in. Describes the automatic regulation of temperature, time, level, flow, and pressure with illustrations and specifications of mechanisms used.

Industrial Uses of Micarta—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Bulletin, 28 pages, 8½x11 in. Pictorial presentation of the uses of Micarta in the construction of radio instruments, electrical parts, aircraft and automobiles; also its applications in plant equipment, paper mills, steel mills, textile mills, chemical and marine installations.

Low Temperature Brazing Alloys—Handy & Harmon, 82 Fulton St., New York, N. Y. Folder, 6x9 in. New Standard sizes for Sil-Fos and Easy-Flo brazing alloys.

Manual for Porcelain Enamel—Porcelain Enamel Institute, Inc., 612 N. Michigan Ave., Chicago, Ill. 24 pages, 8½x11 in. Describes the sources, nature, preparation, and applications of porcelain enamel. Nicely illustrated, supplemented with sales talk to emphasize utility of finish.

Mechanical Rubber Goods—The B. F. Goodrich Co., Akron, Ohio. 27 pages, 8½x11 in. Illustrates and describes rubber industrial products and accessories, belting, hose and fittings, tubing, packing, cements, valves and matting. Contains general and engineering data to guide in making selections and installations.

Mercury Switches—Durakool, Inc., 1012 N. Main St., Elkhart, Ind. Circular No. 300, 6 pages. 8½x11 in. Illustrates operation, construction, sizes and types of metallic envelope mercury switches.

Metal Surface Preparation—E. I. du Pont de Nemours & Co., Wilmington, Del. Booklet, 12 pages, 8½x11 in. A manual for the application of sodium cyanide solutions in the preparation of metal surfaces, with an outline of the various methods used.

Molding with Lumarith—Celluloid Corp., 10 E. 40th St., New York, N. Y. Manual, 24 pages, 6x9 in. Sets forth the nature and properties of Lumarith, a thermoplastic cellulose acetate material. Applications and forms are listed. The advantages and technique of injection molding are presented stressing general principles.

Motorized Speed Reducers—Janette Mfg. Co., 556-558 W. Monroe St., Chicago, Ill. Bulletin 22-10. Folder 3¼ x 6¼ in. Illustrates and describes various types of motors with speed reducers attached ranging from 1/50 to 7½ hp.

Motor Operated Controllers—Automatic Temperature Control Co., 34 E. Logan St., Philadelphia, Pa. Bulletin G-11, 4 pages, 8½x11 in. Describes and illustrates a temperature control mechanism, the type of heating work for which it is suitable, and its operating principles.

Oilless Bronze Bearings—R. W. Rhoades Metaline Co., Inc., P.O. Box No. 1, Long Island City, N. Y. Folder 8½ x 11 in. Short description of construction and operation of "Metaline" bearings. Table of standard sizes is included.

Oil Sealing Rings—The Garlock Packing Co., Palmyra, N. Y. Catalog, 40 pages, 8½x10½ in. General description of construction and types of Garlock Klozures with halftone illustrations. Line cuts show typical applications. Standard sizes are listed.

The Oxygen Lance—The Linde Air Products Co., 205 East 42d St., New York, N. Y. 8 pages, 8½x11 in. Describes the oxygen lance, how it is used and what it will do. Illustrated with diagrams and pictures showing use as a production tool.

Perforated Sheet Metals—Erdle Perforating Co., 167 York St., Rochester, N. Y. Folders, 3½x6 in. Hints on ordering perforated metals with sketch of sheet showing two styles of slots.

Power Transmission Equipment—Boston Gear Works, Inc., Boston, Mass. Catalog 51, 256 pages, 3⅝x6 in. Contains complete information on company's transmission equipment. Illustrated with cuts, data tables, and formulas.

Precision Bearings for Aircraft Controls—Norma-Hoffmann Bearings Corp., Stamford, Conn. Catalog F-956, 24 pages, 5 x 7¼ in. Ball and roller bearings specially designed for applications in aircraft controls. Contains tables of sizes, dimensions and load data, and illustrations of a few heavier types.

Roller, Griplock and Z-Metal Chains—Chain Belt Co., Milwaukee, Wis. Bulletins 291, 292 and 293. Folders each 6x9 in. Construction, applications and advantages are described, with tables of sizes and capacities.

Stock Chain Drives and Couplings—Morse Chain Co., Ithaca, N. Y. 48 pages, 8½x11 in. Contains miscellaneous data in table form on silent chains, roller chains, sprockets and flexible couplings with lists of stock sizes for easy reference.

Thermometers and Pressure Gages—The Brown Instrument Co., Philadelphia, Pa. Catalog No. 6703, 72 pages, 8x10½ in. Covers a complete line of thermometers and pressure gages of the indicating, recording and controlling types, explains constructional features, and gives a detailed description of each instrument.

Variable Speed Transmission—Graham Transmissions, Springfield, Vermont. Bulletin No. 302, 12 pages, 8½x11 in. Featuring the construction, principles of operation, control, and standard types of the Graham Variable Speed Transmissions. Includes dimensional tables with speeds and capacities.

Zinc Base Die Casting Alloy—Apex Smelting Co., 2554 Fillmore St., Chicago, Ill. Metalgram No. 16 and No. 16A, each 4 pages, 8½x11 in. The first of a short series which will show the effect of tin and lead contamination on zinc base die casting alloys Nos. 3 and 5, illustrated with charts.

BOOKS AND BULLETINS

Electron Tubes in Industry

Keith Henney. 539 pages, 6x9 in. More than 400 illustrations. Black clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $5.

The purpose of this second edition is to acquaint industrial engineers with the many applications of amplifier, oscillator, photo-tubes, rectifiers, thyratrons, grid-flow tubes and cathode-ray tubes to non-communication industries.

After an introductory chapter of the fundamentals of electronic tube circuits, with a brief discourse on electron properties, electromotive force, laws, reactance, impedance and resonance, various types of thermionic tubes are discussed. In this chapter much is written and illustrated with line drawings to clear up the mystery, especially to the layman, that surrounds the functions of these tubes.

The third and fourth chapters of over 100 pages each on vacuum-tube amplifiers and gaseous triodes are similarly treated. Light-sensitive tubes and applications of these tubes to industrial installations are covered in two chapters, while rectifiers, cathode-ray tubes and other miscellaneous tubes complete the text.

Throughout the book, the various types of tubes are illustrated with line drawings and photographs. There are many circuit diagrams, charts, tables and detail drawings of methods and industrial applications. Footnotes on many pages and bibliographies at the ends of chapters contain more than 300 references to books and articles on the subjects under discussion More than 400 authors who are quoted in the book are listed in an Author's Index. A subject cross-index of eleven pages completes the book.

Mathematics of Modern Engineering—Vol. I

Robert E. Doherty and *Ernest G. Keller.* 314 pages, 6x9¼ in. 82 line figures. Clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $3.50.

In this volume the authors have done much to show how mathematics can be applied to those branches of physics that lie in the field of engineering. Throughout the book only those aspects of mathematics are dealt with which experience has indicated to be of value to engineers. The only presupposed requisite for understanding this volume is a thorough knowledge of calculus.

After describing the method of mathematical formulation of engineering problems, modern engineering mathematics is studied by showing the underlying idea and purpose in back of differential equations, determinants, Faurier's series, dimensional analysis, vector analysis, vector fields, dyadics, and Heaviside's operational calculus, and the manner of using these principles in the solution of typical case problems.

Mechanical Engineers' Handbook Power Volume

Robert T. Kent. 1252 pages, 5½x8½ in. Numerous illustrations and tables. Flexible atholeather binding. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $5.

This is the first of two volumes prepared for mechanical engineers by Robert T. Kent and a staff of specialists. It forms Volume II of the Wiley Engineering Handbook Series. This new "Kent" is entirely rewritten and rearranged. The overall size of the page is greater. Type faces are larger and consequently very legible. Each section of the book has an identifying number in the page numbering system.

Subject matter is particularly devoted to the specialized field of power and its applications. The main sections deal with air, water, heat, combustion and fuels, steam, steam boiler, steam engine, steam turbine, condensing and cooling equipment, refrigeration and ice-making, heating, ventilating and air-conditioning, internal combustion engines, gas producers, transportation, electric power, power test codes, mathematical tables and index.

The data presented is practical and complete, and is arranged so as to be of great service to the engineer who deals with power.

Symposium on High-Strength Constructional Metals

American Society for Testing Materials. 126 pages, 6x9 in. Illustrations, graphs and tables. Published by the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. Prices: Clothboard cover $1.50; heavy paper cover $1.25.

In this volume are presented papers and discussions which made up a symposium held in conjunction with the seventh regional meeting of American Society for Testing Materials, under the auspices of the Pittsburgh District Committee. Papers cover chemical and physical properties, also manufacturing and fabricating properties of metals and alloys used in buildings, bridges, ships, railway car bodies, automobile bodies, airplane wings, tanks and containers. The principal groups dealt with are alloys of aluminum and magnesium, alloys of copper, alloys of nickel, carbon and low-alloy steels, and corrosion-resisting steels.

Procedure Handbook of Arc Welding Design and Practice

The Lincoln Electric Company. 819 pages, 5¾x9 in. 990 illustrations. Simulated leather cover. Published by the Lincoln Electric Co., 12818 Coit Road, Cleveland, Ohio. Price $1.50.

The present volume is a reissue of the highly valued Procedure Handbook in greatly enlarged form. New data presented includes complete information on the following subjects: Characteristics of the welding generator, selection of type of joint, insurance of fusion welded vessels, welding codes, arc cutting, polarity of welding current, horizontal welds, sheet metal welding, effect of electrode size on welding cost, methods of testing weld metals, 4-6 chrome steel, Monel metal, principles of surfacing by welding, standard shapes, plates and plate girders.

For information concerning present-day arc welding practice, the handbook is a complete reference guide for designers, draftsmen, production managers, steel fabricators, erectors and students of welding.

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Wrought Copper Alloys

Chase Brass & Copper Co., Inc., Waterbury, Conn. Bulletin, 60 pages, 8x10¾ in. Paper covers.

Contains lists of common wrought copper alloys with the name of each and trade names which are sometimes applied, nominal composition, and forms in which they are available. Physical property data are given for a number of tempers to demonstrate range. Reference is also made to the working, machining and welding characteristics of the various alloys.

Isolation of Machine Vibrations

The Korfund Co., Inc., 48-15 Thirty-Second Place, Long Island City, N. Y. Vol. 5, No. 1 "Isolation," 20 pages, 8½x11 in.

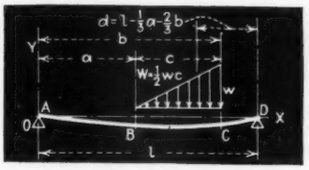
An abstract of a paper presented at a meeting of the American Society of Mechanical Engineers. It deals with the elementary considerations of vibration control, natural cork plates, seismo-dampers using cork, and vibro-dampers using steel springs. Illustrated with photographs and line cuts of installations and mounting details. Engineering and catalog data is added as an aid when making layouts.

Formulas For Beams

I—STATICALLY DETERMINATE CASES

(*Continued from page* 58)

18—END SUPPORTS, PARTIAL TRIANGULAR LOAD



$R_1 = W\,\dfrac{d}{l}$; $R_2 = W\,\dfrac{l-d}{l}$

(A to B), $V = R_1$

(B to C), $V = R_1 - \left(\dfrac{x-a}{c}\right)^2$

(C to D), $V = R_1 - W$

(A to B), $M = R_1 x$; (B to C), $M = R_1 x - W\,\dfrac{(x-a)^3}{3c^2}$

(C to D), $M = R_1 x - \dfrac{1}{3}\,W\,(3x - a - 2b)$;

Max. $M = W\,\dfrac{d}{l}\left(a+\dfrac{2}{3}c\sqrt{\dfrac{d}{l}}\right)$, at $x = a + c\sqrt{\dfrac{d}{l}}$

(A to B), $y = \dfrac{1}{6\,EI}\left[R_1(x^3-l^2x)+Wx\,\dfrac{d^3}{l}+\dfrac{1}{6}c^2\left(1-\dfrac{b}{l}\right)+\dfrac{17}{210}\,\dfrac{c^3}{l}\right]$

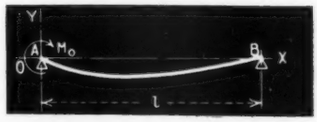
(B to C), $y = \dfrac{1}{6\,EI}\left[R_1(x^3-l^2x) - \dfrac{1}{10}\,W\,\dfrac{(x-a)^5}{c^2} + W\,x\,\left(\dfrac{d^3}{l}+\dfrac{1}{6}\,c^2-\dfrac{1}{6}\,c^2\,\dfrac{b}{l}+\dfrac{17}{270}\,\dfrac{c^3}{l}\right)\right]$

(C to D), $y = \dfrac{1}{6\,EI}\left\{R_1(x^3-l^2x)-W\left[d^3-d^3\,\dfrac{x}{l}-\dfrac{1}{6}\,bc^2\left(1-\dfrac{x}{l}\right)+\dfrac{17}{270}\,c^3\left(1-\dfrac{x}{l}\right)\right]\right\}$

$\theta = \dfrac{1}{6\,EI}\left[-R_1l^2+W\left(\dfrac{d^3}{l}+\dfrac{1}{6}\,c^2+\dfrac{17}{270}\,\dfrac{c^3}{l}-\dfrac{1}{6}\,\dfrac{c^2b}{l}\right)\right]$ at A

$\theta = \dfrac{1}{6\,EI}\left[2\,R_1l^2+W\left(\dfrac{d^3}{l}+\dfrac{17}{270}\,\dfrac{c^3}{l}-\dfrac{1}{6}\,\dfrac{c^2\,b}{l}-3d^2\right)\right]$ at D

19—END SUPPORTS, END COUPLE



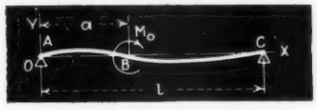
$R_1 = -\,\dfrac{M_0}{l}$; $R_2 = +\,\dfrac{M_0}{l}$; $V = R_1$

$M = M_0 + R_1 x$; Max. $M = M_0$, at A ; $y = \dfrac{1}{6}\,\dfrac{M_0}{EI}\left(3x^2 - \dfrac{x^3}{l} - 2lx\right)$

Max. $y = -\,0.0642\,M_0l^2/EI$, at $x = 0.422\,l$

$\theta = -\,\dfrac{1}{3}\,\dfrac{M_0l}{EI}$, at A ; $\theta = +\,\dfrac{1}{6}\,\dfrac{M_0l}{EI}$, at B

20—END SUPPORTS, INTERMEDIATE COUPLE



$R_1 = -\,\dfrac{M_0}{l}$; $R_2 = +\,\dfrac{M_0}{l}$

(A to C), $V = R_1$

(A to B), $M = R_1 x$; (B to C), $M = R_1 x + M_0$

Max. $-M = R_1 a$, just to left of B ; Max. $+M = R_1 a + M_0$, just to right of B

(A to B), $y = \dfrac{1}{6}\,\dfrac{M_0}{EI}\left[\left(6a - 3\,\dfrac{a^2}{l} - 2l\right)x - \dfrac{x^3}{l}\right]$

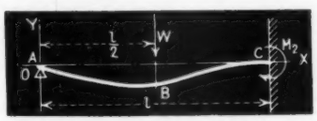
(B to C), $y = \dfrac{1}{6}\,\dfrac{M_0}{EI}\left[3a^2+3x^2-\dfrac{x^3}{l}-\left(2l+3\,\dfrac{a^2}{l}\right)x\right]$

$\theta = -\,\dfrac{1}{6}\,\dfrac{M_0}{EI}\left(2l-6a+3\,\dfrac{a^2}{l}\right)$, at A ; $\theta = +\,\dfrac{1}{6}\,\dfrac{M_0}{EI}\left(l-3\,\dfrac{a^2}{l}\right)$, at C ; $\theta = \dfrac{M_0}{EI}\left(a-\dfrac{a^2}{l}-\dfrac{1}{3}\,l\right)$, at B

II—STATICALLY INDETERMINATE CASES

Loading, Support and Vertical Shear V	Reactions R_1 and R_2 and Constraining Moments M_1 and M_2, Bending Moment M, Max. Positive and Negative Bending Moments, Deflection y, Max. Deflection and End Slope θ

21—ONE END FIXED, ONE END SUPPORTED. CENTER LOAD



$R_1 = \dfrac{5}{16}\,W$; $R_2 = \dfrac{11}{16}\,W$

$M_2 = \dfrac{3}{16}\,W$; (A to B), $V = +\,\dfrac{5}{16}\,W$; (B to C), $V = -\,\dfrac{11}{16}\,W$

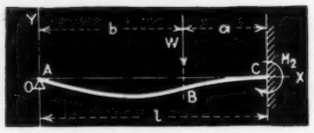
(A to B), $M = \dfrac{5}{16}\,Wx$; (B to C), $M = W\left(\dfrac{1}{2}\,l - \dfrac{11}{16}\,x\right)$

Max. $+M = \dfrac{5}{32}\,Wl$, at B ; Max. $-M = -\,\dfrac{3}{16}\,Wl$, at C

(A to B), $y = \dfrac{1}{96}\,\dfrac{W}{EI}\,(5x^3 - 3l^2x)$; (B to C), $y = \dfrac{1}{96}\,\dfrac{W}{EI}\left[5x^3 - 16\left(x-\dfrac{l}{2}\right)^3 - 3l^2x\right]$

Max. $y = -\,0.00932\,\dfrac{Wl^3}{EI}$, at $x = 0.4472\,l$; $\theta = -\dfrac{1}{32}\,\dfrac{Wl^2}{EI}$, at A

22—ONE END FIXED, ONE END SUPPORTED. INTERMEDIATE LOAD



$$R_1 = \frac{1}{2} W \left(\frac{3\,a^2 l - a^3}{l^3} \right) \; ; \quad R_2 = W - R_1 \; ; \quad M_2 = \frac{1}{2} W \left(\frac{a^3 + 2al^2 - 3a^2 l}{l^2} \right)$$

(A to B), $M = R_1 x$; (B to C), $M = R_1 x - W(x - l + a)$

Max. $+ M = R_1 (l - a)$ at B ; max. possible value $= 0.174\, Wl$ when $a = 0.634l$

Max. $- M = - M_2$ at C ; max. possible value $= - 0.1927\, Wl$ when $a = 0.4227l$

(A to B), $V = R_1$

(B to C), $V = R_1 - W$

(A to B), $y = \frac{1}{6\,EI} \left[R_1 (x^3 - 3l^2 x) + 3Wa^2 x \right]$

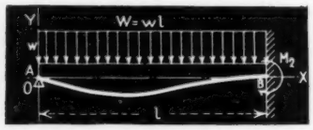
(B to C), $y = \frac{1}{6\,EI} \left\{ R_1 (x^3 - 3l^2 x) + W \left[3a^2 x - (x-b)^3 \right] \right\}$

If $a < 0.586l$, Max. y is between A and B at: $x = l \sqrt{1 - \frac{2l}{3l - a}}$

If $a > 0.586l$, Max. y is at: $x = \dfrac{-2bl^3 + \sqrt{4b^2 l^6 + l^3 (3a^2 l - a^3 - 2l^3)\left(3a^2 - \frac{a^3}{l} + l^2 - 2al\right)}}{3a^2 l - a^3 - 2l^3}$

If $a = 0.586l$, Max. y is at B and $= - 0.0098 \frac{Wl^3}{EI}$, max. possible deflection ; $\theta \parallel \frac{1}{4} \frac{W}{EI} \left(\frac{a^3}{l} - a^2 \right)$, at A

23—ONE END FIXED, ONE END SUPPORTED. UNIFORM LOAD



$$R_1 = \frac{3}{8} W \; ; \quad R_2 = \frac{5}{8} W \; ; \quad M^2 = \frac{1}{8} Wl \; ; \quad M = W \left(\frac{3}{8} x - \frac{1}{2} \frac{x^2}{l} \right)$$

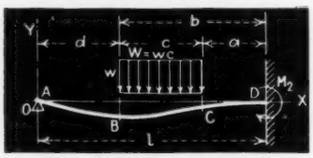
Max. $+ M = \frac{9}{128} Wl$ at $x = \frac{3}{8} l$; Max. $- M = - \frac{1}{8} Wl$, at B

$V = W \left(\frac{3}{8} - \frac{x}{l} \right)$

$$y = \frac{1}{48} \frac{W}{EIl} \left(3lx^3 - 2x^4 - l^3 x \right)$$

Max. $y = 0.0054 \frac{Wl^3}{EI}$, at $x = 0.4215l$; $\theta = - \frac{1}{48} \frac{Wl^2}{EI}$ at A

24—ONE END FIXED, ONE END SUPPORTED. PARTIAL UNIFORM LOAD



$$R_1 = \frac{1}{8} \frac{W}{l^3} \left[4l (a^2 + ab + b^2) - a^3 - ab^2 - a^2 b - b^3 \right] \; ; \quad R_2 = W - R_1 \; ;$$

$$M_2 = R_1 l - \frac{1}{2} W (a + b) \; ; \quad \text{(A to B)}, \; M = R_1 x \; ; \quad \text{(B to C)}, \; M = R_1 x - W \frac{(x-d)^2}{2c}$$

(A to B), $V = R_1$

(C to D), $M = R_1 x - W \left(x - d - \frac{1}{2} c \right)$

(B to C), $V = R_1 - W \left(\frac{x-d}{c} \right)$

Max. $+ M = R_1 \left(d + \frac{1}{2} \frac{R_1}{W} c \right)$, at $x = d + \frac{R_1}{W} c$; Max. $- M = - M_2$

(C to D), $V = R_1 - W$

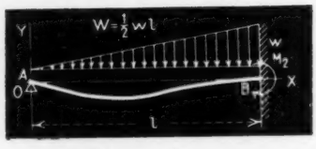
(A to B), $y = \frac{1}{EI} \left[R_1 \left(\frac{1}{6} x^3 - \frac{1}{2} l^2 x \right) + Wx \left(\frac{1}{2} a^2 + \frac{1}{2} ac + \frac{1}{24} c^2 \right) \right]$

(B to C), $y = \frac{1}{EI} \left[R_1 \left(\frac{1}{6} x^3 - \frac{1}{2} l^2 x \right) + Wx \left(\frac{1}{2} a^2 + \frac{1}{2} ac + \frac{1}{6} c^2 \right) - W \frac{(x-d)^4}{24c} \right]$

(C to D), $y = \frac{1}{EI} \left\{ R_1 \left(\frac{1}{6} x^3 - \frac{1}{2} l^2 x + \frac{1}{3} l^3 \right) + W \left[\frac{1}{6} \left(a + \frac{1}{2} c \right)^3 - \frac{1}{2} \left(a + \frac{1}{2} c \right)^2 l - \frac{1}{6} \left(x - d - \frac{1}{2} c \right)^3 + \frac{1}{2} \left(a + \frac{1}{2} c \right)^2 x \right] \right\}$

$\theta = - \frac{1}{EI} \left[\frac{1}{2} R_1 l^2 - W \left(\frac{1}{2} a^2 + \frac{1}{2} ac + \frac{1}{6} c^2 \right) \right]$, at A

25—ONE END FIXED, ONE END SUPPORTED. TRIANGULAR LOAD



$$R_1 = \frac{1}{5} W \; ; \quad R_2 = \frac{4}{5} W \; ; \quad M_2 = \frac{2}{15} Wl$$

$$M = W \left(\frac{1}{5} x - \frac{1}{3} \frac{x^3}{l^2} \right) \; ; \quad \text{Max.} + M = 0.06\, Wl \text{ at } x = 0.4474l \; ; \quad \text{Max.} - M = - M_2$$

$V = W \left(\frac{1}{5} - \frac{x^2}{l^2} \right)$

$$y = \frac{1}{60} \frac{W}{EIl} \left(2lx^3 - l^3 x - \frac{x^5}{l} \right) \; ; \quad \text{Max. } y = - 0.00477 \frac{Wl^3}{EI}, \text{ at } x = l \sqrt{\frac{1}{5}} \; ;$$

$\theta = - \frac{1}{60} \frac{Wl^2}{EI}$ at A

(Continued in March issue)